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AN AUTOMATIC SPEED CONTROLLER
FOR AUTOMOBILES

A THESIS

Presented to

The Faculty of the Graduate Division

by

Ernest Robert Holloway, Jr.

In Partial Fulfillment

of the Requirements for the Degree

Master of Science in Mechanical Engineering

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AN AUTOMATIC SPEED CONTROLLER

FOR AUTOMOBILES

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SUMMARY

The purpose of the reported investigation was to analyze, develop, and test a working model of a vehicle speed controller based on a principle of operation that apparently has not been utilized to date.

When the controller is engaged, the road speed of the vehicle is maintained by varying the accelerator pedal return spring force which in turn controls the accelerator pedal position while a slight foot pressure is applied to the pedal. When the controller is disengaged, the operation of the vehicle is returned to normal.

A literature survey and patent search were made and it was found that there are three different speed control systems which are being used today by the automobile manufacturers. All of these systems are similar in the respect that they control road speed by adjusting the carburetor directly. The proposed system is believed to offer advantages from the standpoint of safety and lower manufacturing cost.

Both an analytical and an experimental investigation were made of the proposed system. A 1964 Oldsmobile furnished the test car model for the speed controller. A mathematical model was determined from acceleration and deceleration data which was obtained by applying step changes in the accelerator pedal position and recording the velocity response of the test car versus time. All of the responses from these tests closely resembled the step input response of a heavily overdamped second-order system.

A laboratory experimental model was constructed which incorporated

an actual automobile accelerator pedal and linkage coupled to an analog computer on which the test automobile dynamics were simulated by a heavily overdamped second-order system. The speed control mechanism was mounted on the simulated firewall along with the automobile linkage parts.

The proposed speed control system was mathematically analyzed by the use of the describing function technique and Nyquist's criterion. This analysis indicated that although the system was stable, it tended to limit cycle. Therefore, three alternate approaches were examined. They were: 1) tolerate the limit cycles, 2) adjust the system gain or the deadband to eliminate the limit cycles, or 3) compensate the system in some manner, possibly by adding derivative control, to eliminate the limit cycles.

Experimental tests were run to verify the predictions of the mathematical analysis. The tests verified the existence of limit cycles and showed the movement of the accelerator pedal during limit cycling was too excessive to tolerate. Attempts to eliminate the limit cycles by adjusting the system gain were unsuccessful. Although enlarging the deadband eliminated the limit cycles, it was an impractical adjustment because it decreased the accuracy of the system. The addition of derivative control to compensate the system completely eliminated the limit cycles.

A series of tests were run to evaluate the system operation when the system parameters were varied. The control system was tested at both high and low vehicle speeds and found to operate well for all cases. Although the system worked for both large and small deadbands, the smaller deadbands produced the most accurate speed control. It was determined that a control spring with a constant of 7 lbs./in. or more, allowed the system to function properly for both a light and a heavy foot pressure on the pedal. Also, the system response to road disturbances such as an increasing or decreasing road slope was tested and found to be very good.

Based on the results of all the tests that were run, the proposed control system is an excellent device for controlling the road speed of automobiles.

NOMENCLATURE

C	system output
d	width of deadband, mph; volts
e_i	error signal to relays, mph; volts
e_o	output of relays, volts
j	imaginary unit
K	constant
R	system input
S	laplace operator
t	time, seconds
V	automobile velocity, mph; volts
V_D	automobile velocity at which the relays are set to throw, mph; volts
V_R	operating point velocity, mph; volts
V_t	automobile velocity at which the relays throw, mph; volts
V_m	motor voltage, volts
V_r	output velocity, mph; volts
$\dot{V}$	automobile acceleration, miles/hr^2; volts
ω	frequency, radians/sec.
ω_c	crossover frequency, radians/sec.
ω_n	natural frequency, radians/sec.
ζ	damping ratio

CHAPTER I

INTRODUCTION

The objective of the study reported in this thesis was to analyze, develop, and test a working model of a vehicle speed controller based on a principle of operation that apparently has not been utilized to date.

Ever since the development of the automobile as a reliable mode of transportation, there has been a continuous effort by the automotive industry to improve its controllability. Many improvements such as power brakes, automatic transmissions, and power steering have been developed to reduce driver effort and increase the pleasure of driving. One of these improvements has been the development of automatic speed controlling devices for automobiles.

Speed controlling devices for automobiles may be broken down into two basic categories: 1) road speed limiters, and 2) road speed controllers. Road speed limiters often referred to as road speed governors have been in existence since the early development of the automobile. These governors maintain a constant throttle opening position and may utilize any type of engage and override features. The inability of these governors to permit a sudden acceleration past the maximum limited speed to pass another vehicle has severely restricted their application to the automotive industry. Also the inability of these limiters to successfully adjust to changes in load such as an increasing or decreasing road slope has hampered their desirability.

In contrast road speed controllers operate as closed-loop servomechanisms and therefore have the ability to adjust to changing road conditions. These devices maintain a constant road speed by comparing the actual speed of the automobile with the desired speed and then adjusting the actual speed by an amount proportional to the difference between the actual and desired speeds. As a result of this ability to adjust to changing conditions and the advantage of accelerating beyond the control speed, the road speed controllers are more desirable for automobile application than the road speed limiters.

CHAPTER II

HISTORY AND LITERATURE SURVEY OF PREVIOUS WORK

The two factors which have had the greatest influence on the development of automatic road speed controllers are need and technology. As with the majority of new inventions a need had to be created before an idea could be conceived. In the early days of automobile travel the highways were unsuited for constant speed driving and therefore the need for an automatic road speed controller was relatively nonexistent. However, with the development of better highways and the increasing mileage of limited access highways the need for an automobile speed controller became apparent.

Once the need had been created, development was dependent mainly on available technology to formulate an idea and produce a working control device.

The field of automatic controls had taken great strides and gained much experience during World War II. In the years following the War the field continued to expand as the experience and knowledge of automatic controls spread throughout industry. Eventually, the need for an automobile speed controller was answered by a basic concept of automatic controls; the closed-loop servomechanism system.

Since the introduction of the first automatic vehicle speed controller in the mid 1950's, many inventions and modifications of vehicle speed controllers have been patented. All of those controllers operate as closed-loop servomechanisms to position the throttle an amount proportional to the difference between the actual and desired speeds. The majority of these controllers utilize engine intake manifold vacuum acting on a diaphram to position the throttle while other controllers utilize electric motors or hydraulic pistons as prime movers.

The following is a brief discussion of the principle of operation of the three most representative and widely used vehicle speed controllers.

The initial development of an automatic vehicle speed controller was by the Perfect Circle Corporation, Hagerstown, Indiana in 1957 (1). The mechanism, called the Speedostat, is an electromechanical unit driven by the transmission through a cable. When a desired speed is set on a selector dial, located inside the vehicle, a cable from the dial sets a calibrated spring inside the unit mounted under the hood. This spring controls a small flyball governor which opens or closes contacts energizing a small 12-volt motor. The motor, running forward or in reverse according to the action of the governor, actuates a rocker arm which positions the throttle opening. For long periods of constant speed driving a locking feature is provided to maintain the desired speed without need to hold down the accelerator pedal. Acceleration past the set speed is possible at any time and a touch of the break pedal immediately overrides the system.

In 1963 Rex Industries developed a "cruise control" which uses engine manifold vacuum to provide power for the system (2). When the cruising speed is set on the dashboard dial, a cable from the dial sets the limiting amount of air pressure that can be produced by a governor which is driven by the speedometer cable. Centrifugally actuated weights in the governor regulate the flow of air to a pneumatic servo control valve. The control valve regulates the amount of intake manifold vacuum which is applied to the spring opposed diaphragm throttle actuator. A pedal valve is a safety feature which will bleed air to the intake manifold unless a slight foot pressure is exerted on the brake pedal. The driver may override the system at anytime.

Another automatic speed control device which uses engine manifold vacuum to power the system was developed by the AC Spark Plug Division of General Motors (3). Called the Electro-Cruise, this system also uses a diaphragm throttle positioner which receives its motivating power from the engine manifold vacuum. However, the diaphragm throttle positioner is controlled by a solenoid valve. The solenoid valve is operated by electrical signals representing the difference between the desired speed and the actual road speed. The electrical signals are generated by a speed transducer located in the speedometer mechanism. The driver may accelerate past the cruising speed at any time and the system may be disengaged by depressing the break pedal, pulling out the engage knob, or turning off the ignition switch.

Since the introduction of the first automatic speed controller in 1957, the device has increased in popularity with each year. At the present time all of the major automobile manufacturers in the United States offer at least one of the three previously mentioned controllers. Over the years each automobile manufacturer has made various minor modifications to the systems such as different types and locations of engage and cruise speed select knobs. However, the basic principle of operation of the three speed controllers has remained the same as previously described.

CHAPTER III

THEORETICAL INVESTIGATION

Principle of Operation

The accelerator pedal on the majority of automobiles is returned to its off position by the force of a retaining spring. This spring is attached to the engine throttle linkage and is usually located on the engine side of the firewall.

On the subject automobile road speed controller this retainer spring is replaced by two springs: 1) a stationary spring and 2) a control spring. The stationary spring will be engaged at all times while the control spring will be partially or fully engaged or disengaged depending on the speed of the vehicle. The stationary spring has a relatively small spring constant so that the accelerator pedal can be depressed easily with a small foot pressure when the control spring is disengaged. However, under the same condition, the stationary spring must be strong enough to return the accelerator pedal to its off position when the foot force is released. The control spring has a larger spring constant to enable it to move the accelerator pedal against the opposition of a constant foot force.

The engagement and disengagement of the control spring is accomplished by a small electric motor. The motor will be turned on and off by a pair of electrical contacts which are located on the speedometer an increment, ΔV, above and below the desired cruising speed. The desired

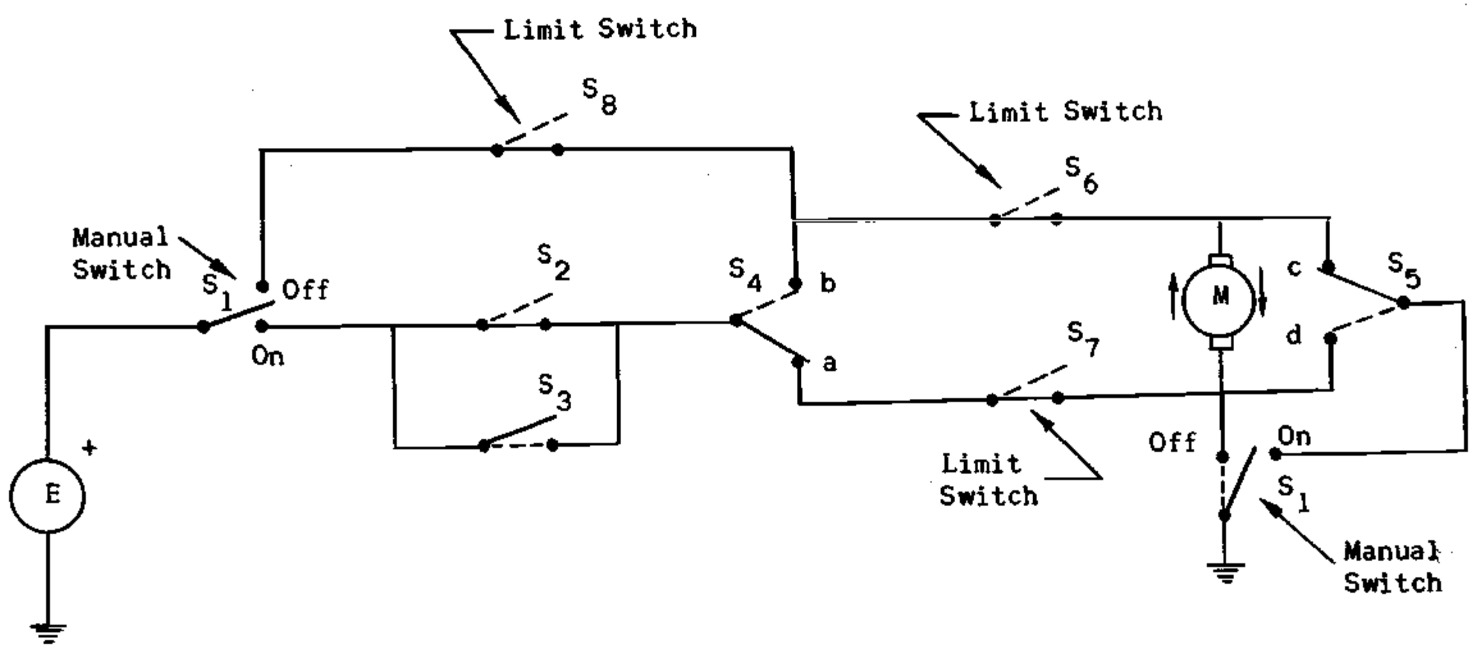

Figure 1. Control System Electrical Circuit Diagram.

cruising speed is manually set by means of a knob located on the dashboard of the vehicle.

The electrical circuit diagram of the control system is shown in Figure 1. The system is turned on by a manual switch S_1 which is located on the dashboard. Current immediately passes through the circuit energizing the motor which disengages the control spring. This allows a slight foot force, which is now only opposed by the stationary spring, to accelerate the vehicle easily. When speed V_1 is reached (see Figure 2) the lower contact energizes a relay which opens switch S_2 and throws switch S_4 to position b, cutting the motor off.

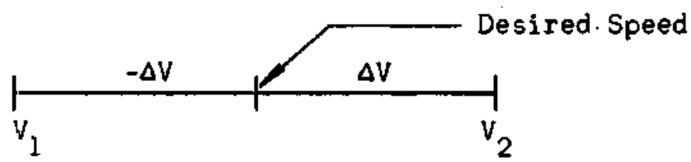

Figure 2. Cruising Speed Deadband Range.

However, if the speed continues to increase through the deadband until V_2 is reached, the upper contact energizes a relay which closes switch S_3 and throws switch S_5 to position d. Current will now pass through the circuit causing the motor to engage the control spring which in turn causes the vehicle to decelerate. When the speed drops below V_2, the upper contact deenergizes the relay causing S_5 to return to position c and S_3 to open. This action stops the motor and holds the control spring in a fixed position until the vehicle speed closes either the upper or the lower contact.

The switches S_6 and S_7 are used to limit the end travel of the control spring. When S_1 is thrown to the off position, the motor engages the control spring until limit switch S_8 opens. This partial engagement of the control spring combined with the stationary spring is equivalent to the standard spring which is designed for the vehicle. Therefore, with the system turned off the operation of the accelerator pedal is returned to normal.

When the system is in operation, the total effect is that the accelerator pedal will reach an equilibrium position that will cause the vehicle speed to remain within the desired deadband range.

Theoretical Analysis

The principal objective in making an analytical analysis is to investigate the stability of the proposed control system. Since the system contains a nonlinear component (the relays with deadband), the most meaningful analysis is one utilizing describing-function techniques. In making this analysis an "approximate transfer function," $G_D(j\omega)$, was chosen for the nonlinear component and stability was checked by application of Nyquist's criteria (4) and the describing function techniques (5) to the system's characteristic equation

$$1 + G_1(j\omega)\, G_D(j\omega) = 0 \tag{3.1}$$

where $G_1(j\omega)$ is the sinusoidal transfer function of the system's linear components.

The complete speed control system block diagram is shown in Figure 3.

An assumption was made in arriving at the block diagram of Figure 3 that the accelerator pedal was positioned directly by the control

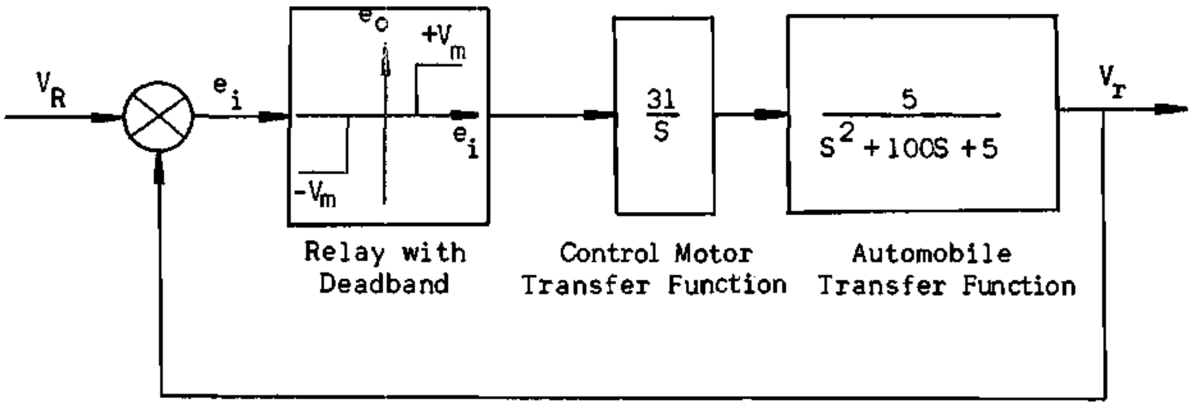

Figure 3. Speed Control System Block Diagram.

motor. This was found to closely approximate the actual system in operation anytime a nominal foot pressure was applied to the accelerator pedal. A nominal foot pressure would be one that would override the stationary spring but not completely override the combined stationary and control springs. The determination of the control motor transfer function and the automobile transfer function are discussed in detail in Chapter IV and Appendix III.

$G_1(S)$, the transfer function of the system's linear components, is determined by combining the control motor and automobile transfer functions

$$G_1(S) = \frac{155}{S(S^2 + 100S + 5)} \tag{3.2}$$

The linear component sinusoidal transfer function is obtained by replacing S in equation 3.2 by $j\omega$ to give

$$G_1(j\omega) = \frac{155}{j\omega(-\omega^2 + 100j\omega + 5)} \tag{3.3}$$

The input-output curve and the describing function for the relay with deadband are given by Figure 4 and the following equations, (3.4) and (3.5),

$$G_D(j\omega) = \frac{4V_m}{\pi X}\sqrt{1 - R^2}\ \angle 0^\circ \qquad X > \frac{d}{2} \tag{3.4}$$

$$G_D(j\omega) = 0 \qquad X < \frac{d}{2}$$

$$R \triangleq \frac{d}{2X} \tag{3.5}$$

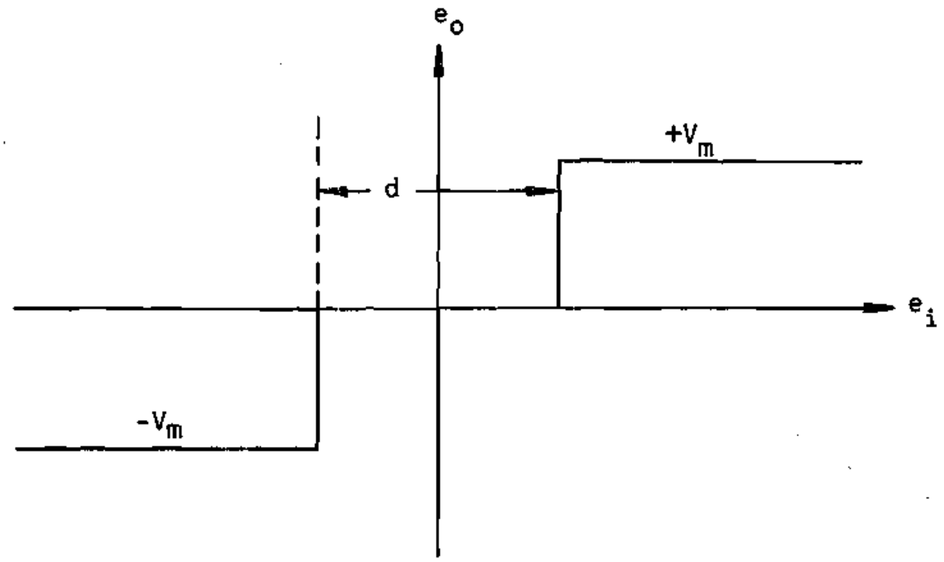

Figure 4. Input-Output Curve for Relay with Deadband.

The defining equations for the describing function are

$$e_i = X \sin \omega t \tag{3.6}$$

$$e_o = 0 \qquad 0 \leqq \omega t \leqq \alpha \tag{3.7}$$

$$e_o = + V_m \qquad \alpha \leqq \omega t \leqq \pi - \alpha \tag{3.8}$$

$$e_o = 0 \qquad \pi - \alpha \leqq \omega t \leqq \pi + \alpha \tag{3.9}$$

$$e_o = - V_m \qquad \pi + \alpha \leqq \omega t \leqq 2\pi - \alpha \tag{3.10}$$

$$e_o = 0 \qquad 2\pi - \alpha \leqq \omega t \leqq 2\pi \tag{3.11}$$

$$\alpha = \sin^{-1} \frac{d}{2X} \tag{3.12}$$

Since this describing function introduces no phase angle, it may be considered as a variable gain.

Before proceeding with the analysis it should be noted that the assumptions necessary for the validity of the describing function technique are as follows: 1) the system is unforced and time invariant, and 2) the linear transfer function contains sufficient low-pass filtering to allow excluding from consideration the harmonics in the output of the nonlinearity (6).

The application of the describing function technique calls for Nyquist plots of $G_1(j\omega)$ and $\frac{-1}{G_D(j\omega)}$. A limit cycle is predicted if

$$G_1(j\omega) = \frac{-1}{G_D(j\omega)} \tag{3.13}$$

A limit cycle is the existence of sustained oscillations at the frequency for which equation (3.13) is satisfied.

According to the Nyquist criterion, stability is determined by the enclosure of any point on the $\frac{-1}{G_D(j\omega)}$ curve by the $G_1(j\omega)$ curve providing there are no poles in the right half of the s-plane. If $G_1(j\omega)$

contains no poles in the right half of the s-plane, as is the case here, then it is only necessary to sketch the Nyquist plot of $G_1(j\omega)$ that corresponds to $\omega = \infty$ to $\omega = 0$ on the Nyquist path to examine enclosure. Plots of $G_1(j\omega)$ and $\frac{-1}{G_D(j\omega)}$ are shown in Figure 5 and by inspection it is seen that a possibility of the two curves intersecting definitely exists.

The frequency, ω_c, at which the Nyquist plot of $G_1(j\omega)$ intersects the negative real axis of the GH plane, called the crossover frequency, is found by setting the imaginary part of $G_1(j\omega)$ equal to zero.

$$5 - \omega^2 = 0 \tag{3.14}$$

$$\omega = \omega_c = \sqrt{5} \tag{3.15}$$

The magnitude of $G_1(j\omega)$ as it crosses the real axis is found by substituting equation (3.15) into equation (3.3) and is -.31. Therefore, the -1 point is not enclosed and by Nyquist's criterion the linear portion of the system is stable. The crossover point is shown in Figure 5.

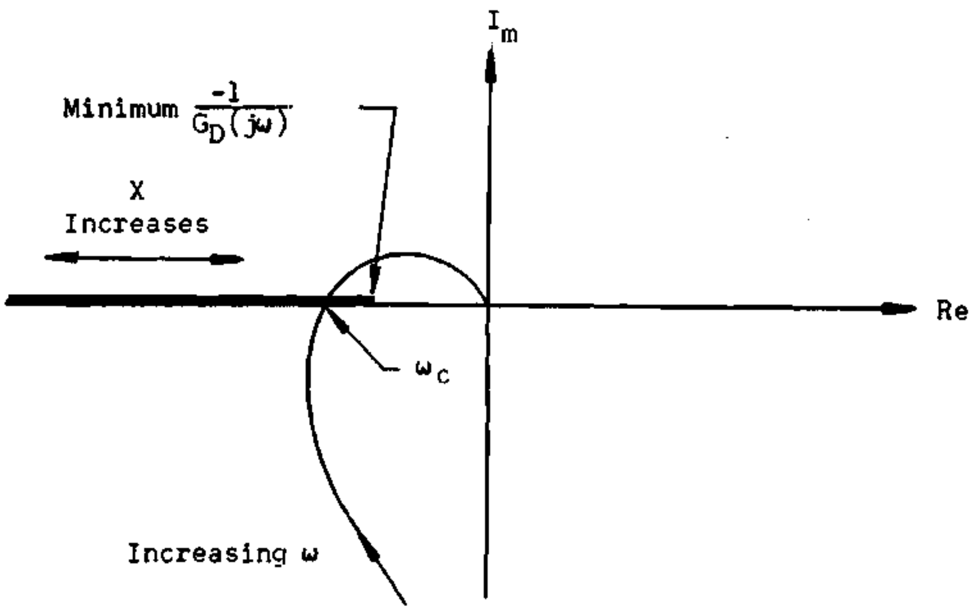

Figure 5. Nyquist Diagram of $G_1(j\omega)$ and Plot of $\frac{-1}{G_D(j\omega)}$.

In order to plot the inverse describing function curve $\frac{-1}{G_D(j\omega)}$ it is necessary to closely examine the value of the describing function as the input X goes from zero to infinity. A sketch of $G_D(j\omega)$ versus input is shown in Figure 6 for any given and physically possible deadband, d, and output voltage V_m.

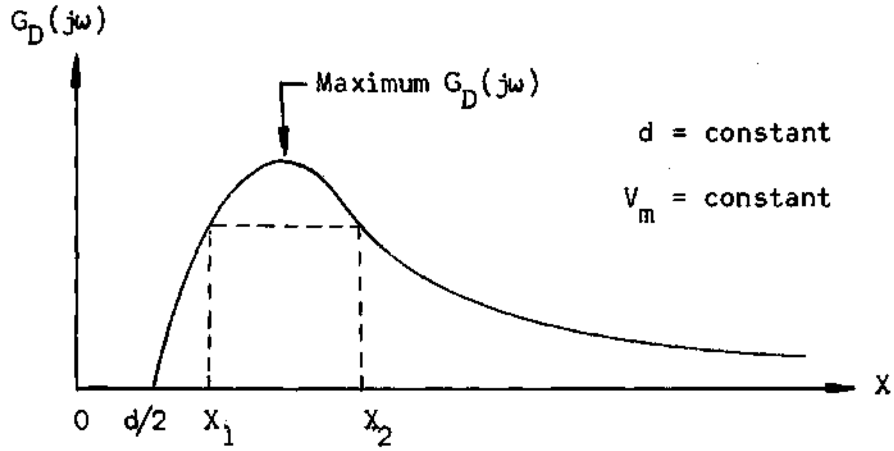

Figure 6. Sketch of $G_D(j\omega)$ versus Input X.

For the describing function to have an output, the input magnitude, X, must have a value greater than $\frac{d}{2}$. If X is forced to range from this value upward, $\frac{-1}{G_D(j\omega)}$ will decrease and approach zero from $-\infty$ along the real axis. It will reach a minimum and then return once again to $-\infty$ along the real axis. Now, assuming that the nonlinear component only receives sinusoidal signals from the linear components of the system, stability and limit cycle characteristics can be determined by assuming values of input magnitude corresponding to various points along the $\frac{-1}{G_D(j\omega)}$ plot.

Let X_1 and X_2 shown in Figure 6 be the two values of input at the intersections of the $\frac{-1}{G_D(j\omega)}$ and $G_1(j\omega)$ loci. If $X < X_1$, the system is stable and oscillations will decay. However, if $X > X_1$, the system will be unstable and the magnitude of the oscillations will increase, causing the magnitude of $\frac{-1}{G_D(j\omega)}$ to reach a minimum before again increasing until $X = X_2$. Oscillations will continue indefinitely at this magnitude, since any increase would cause the system to become stable and result in a decay of the magnitude.

Since the stability analysis of the proposed speed control system has indicated the existence of limit cycles, three alternatives to the system development are left. The choices are: 1) live with the limit cycles, 2) adjust the system gain or $G_D(j\omega)$ so that the loci of $G_1(j\omega)$ and $\frac{-1}{G_D(j\omega)}$, shown in Figure 5, will never intersect, or 3) compensate the system in some manner to eliminate the limit cycles.

Economically speaking, living with the limit cycles would be the best solution because it would require no additional system components. However, whether living with the limit cycles is feasible or not can best be determined by experimentation which is discussed in the following chapter.

Assuming it is not feasible to tolerate the limit cycles, one way to eliminate them would be to adjust the system gain or $G_D(j\omega)$ so that the loci of $G_1(j\omega)$ and $\frac{-1}{G_D(j\omega)}$ will never intersect. Since the gain of the automobile transfer function of Figure 3 cannot be changed, the only way to alter the linear system gain would be to adjust the control motor gain. Reducing this gain would reduce the value at which the Nyquist plot of $G_1(j\omega)$ crosses the negative real axis and possibly prevent the intersection of the two loci shown in Figure 5. However, reducing the gain of the control motor would mean slowing the adjustment of the accelerator pedal. This would cause the system to be slow responding and sluggish and would therefore be a disadvantage. An attempt will be made in the experimental investigation to eliminate the limit cycles by reducing the control motor gain.

Another way to prevent an intersection of the two loci would be to reduce the maximum value of $G_D(j\omega)$ so that the minimum point of the inverse describing function curve $\frac{-1}{G_D(j\omega)}$ would not be enclosed by the plot of $G_1(j\omega)$. The most effective way to reduce the maximum value of $G_D(j\omega)$ is to increase the size of the deadband. However, increasing the deadband would also be a disadvantage because the probability of settling at and maintaining the desired cruising speed decreases as the deadband increases.

To calculate the minimum deadband size (for the system shown in Figure 3) which will allow no intersection of the $G_1(j\omega)$ and $\frac{-1}{G_D(j\omega)}$ loci, the first step was to differentiate equation (3.4) with respect to X and set it equal to zero. By doing this the maximum value of $G_D(j\omega)$ was found to occur when

$$X = .707d \tag{3.16}$$

At the intersection of the $G_1(j\omega)$ and $\frac{-1}{G_D(j\omega)}$ loci, $G_D(j\omega)$ has a value of 3.23. Therefore, by setting the right hand side of equation (3.4) equal to 3.23 and by substituting equation (3.16) into equation (3.4) the minimum deadband value for no loci intersection was found to be 4.73 mph. In effect this means that for deadbands greater than 4.73 mph there will be no loci intersection and limit cycles cannot occur.

A similar calculation can be made for the system when the control motor gain has been reduced. When the control motor gain is reduced by a factor of 5, the magnitude of $G_1(j\omega)$ as it crosses the real axis is -.062. Therefore, for no intersection of $G_1(j\omega)$ and $\frac{-1}{G_D(j\omega)}$ the maximum value of $G_D(j\omega)$ should be less than 16.2. In effect, this means that for deadbands greater than 1.2 mph there will be no loci intersection and limit cycles cannot occur. Tests will be run in the experimental investigation to determine if the above calculations agree with the actual system operation.

Probably the best way to eliminate the limit cycles would be to bring the entire Nyquist plot of $G_1(j\omega)$ below the negative real axis. This would mean that regardless of the value of the inverse describing function curve, the Nyquist plot would never intersect it. As a result, no limit cycles could exist.

The Nyquist plot of $G_1(j\omega)$ can be brought below the negative real axis by adding derivative control to the system as shown in Figure 7.

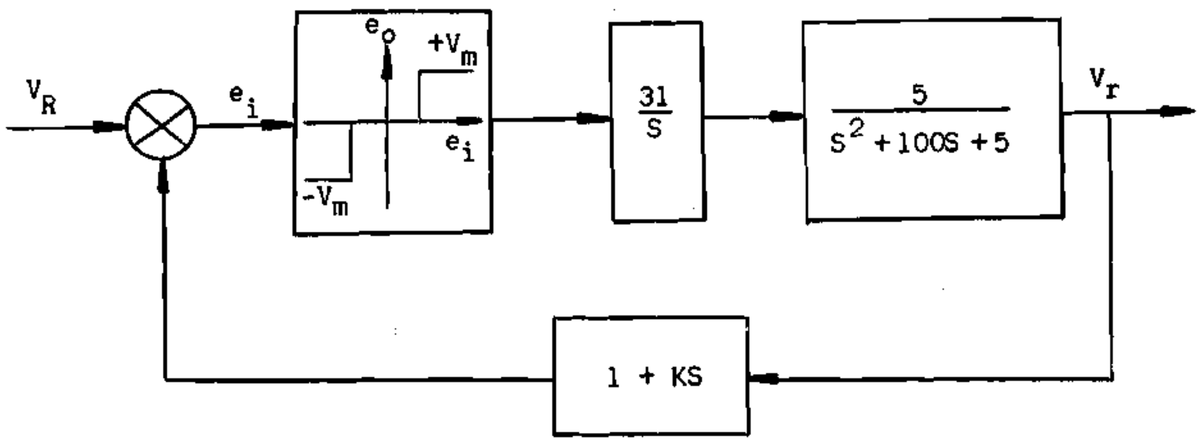

Figure 7. Speed Control System Block Diagram with Derivative Control.

The linear loop transfer function is now written

$$G_1(S) = \frac{155(1 + KS)}{S(S^2 + 100S + 5)} \tag{3.17}$$

and its Nyquist plot is shown in Figure 8. There is no intersection of the two loci possible, and thus, no limit cycle can exist.

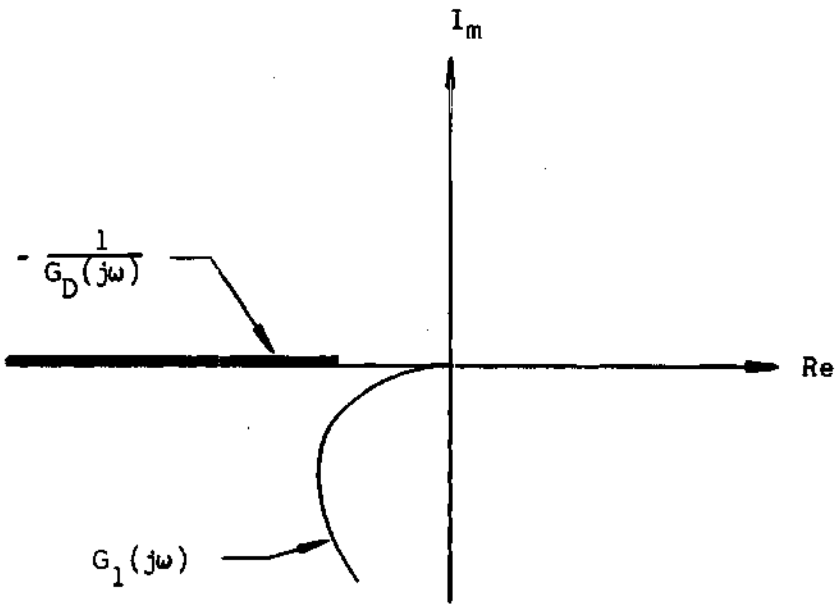

Figure 8. Nyquist Diagram of Compensated System and Plot of $\frac{-1}{G_D(j\omega)}$.

CHAPTER IV

EXPERIMENTAL INVESTIGATION

Objectives

The principal objectives in the experimental investigation were to: 1) verify the existence of limit cycles as predicted by the theoretical analysis, 2) determine the feasibility of tolerating the limit cycles, 3) determine the effect of reducing the control motor gain and increasing the size of the deadband, 4) examine the system behavior with the addition of derivative control, and 5) evaluate the system performance by varying the parameters of the system such as the control spring constant, the size of the deadband, the cruising speed, etc.

Experimental Equipment

The experimental equipment arrangement is shown in Figure 9.

The main test stand was composed of standard Oldsmobile throttle linkage parts and a linear potentiometer mounted on a simulated wooden firewall.

The analog computer was used to simulate an experimentally determined mathematical model for the automobile. The comparator relays on the computer were used to control the power supply to the motor. Also, the computer power supply and one of its potentiometers were utilized to simulate a change in road slope or other disturbance.

A 28 volt d.c. control motor and gear box, mounted on the wooden firewall, were used to rotate the actuator drum which engaged and

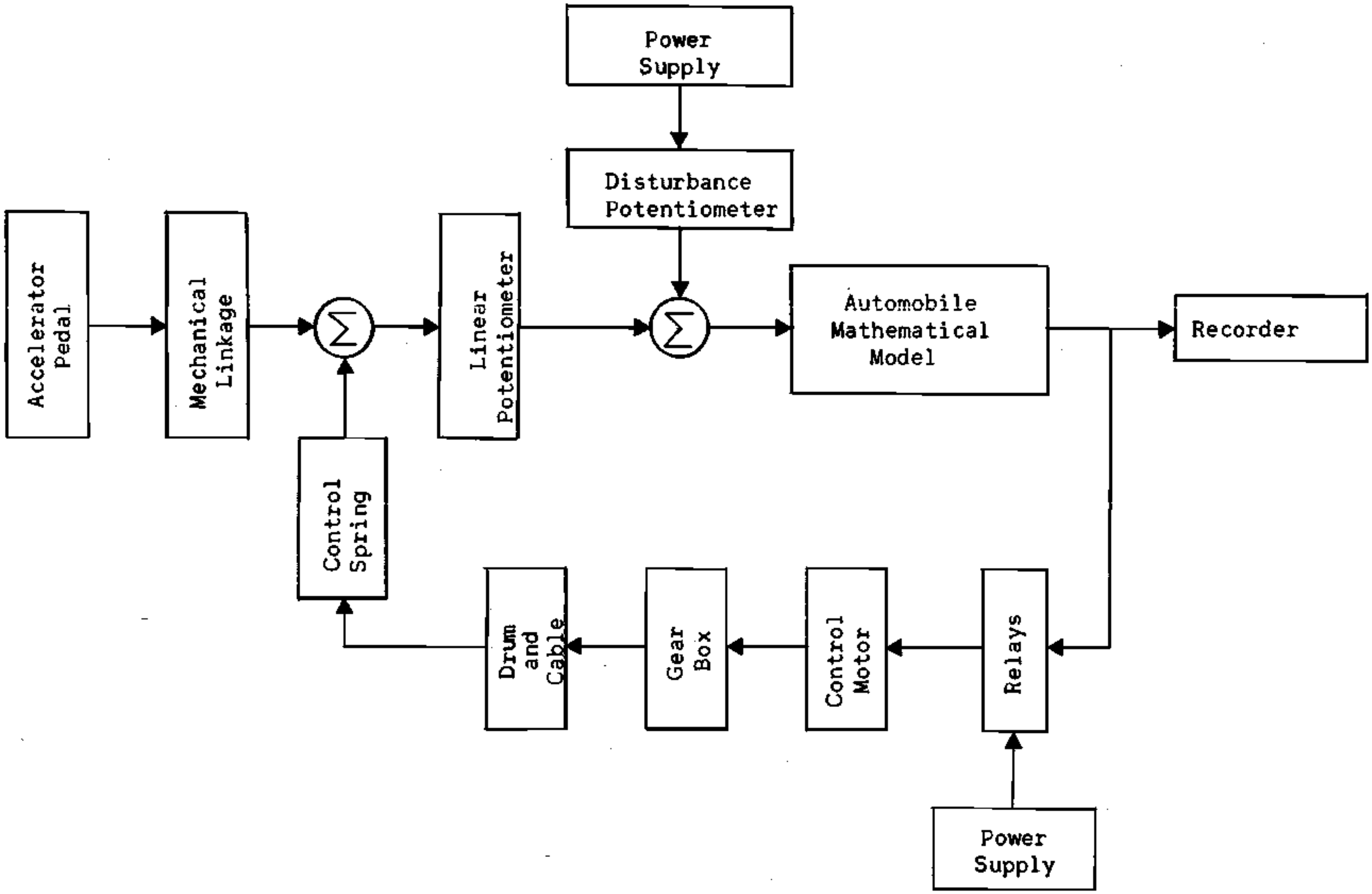

Figure 9. Block Diagram of Experimental Setup.

disengaged the control spring. The 28 volt d.c. control motor was used with a reduced voltage on many tests and found to operate satisfactorily down to 8 volts.

A detailed description of the experimental equipment used is given in Appendix II.

General Procedure

The initial task in the experimental investigation was to secure data which would describe the speed response of the test vehicle. A 1964 Oldsmobile with V-8 engine and automatic transmission was used for the tests. A reasonably level stretch of road on U. S. Highway I-285 at the intersection of U. S. Highway I-85 was used for the test course. All tests were run traversing the course of approximately 1.4 miles in the same direction. Data composed of vehicle speed versus time was taken for varying step inputs to the throttle linkage. The step inputs which varied from 3/64 in. to 3/16 in. were taken from operating points of 30, 45, and 55 mph. Four steps were taken at each operating point--two for acceleration and two for deceleration. Since the purpose of the speed controller is to hold a constant highway speed, it was felt that taking data at the indicated operating points was more representative than taking data from an initial velocity of zero. The results of these acceleration and deceleration tests are given in Appendix III along with the development of the mathematical model for the automobile.

Once the mathematical model for the automobile had been simulated on the analog computer, the laboratory system was complete and ready for testing.

The first laboratory test was to verify the existence of limit cycles in the system without derivative control. A weight of 5 lbs. was hung on the accelerator pedal linkage to simulate a constant force on the accelerator pedal and the deadband was set for 53 to 57 mph.

A test was run to find out if the limit cycling would be tolerable. A deadband of 2 mph was used because a velocity fluctuation of more than 2 mph would be undesirable for long periods of operation. The movement of the accelerator pedal was recorded as the system limit cycled.

Other tests were run to determine if the limit cycles could be eliminated by reducing the control motor gain and increasing the size of the deadband. The control motor gain was decreased by reducing the speed of rotation of the output actuator shaft. This was accomplished by adding an auxiliary shaft with a reduction of 5 to 1.

Tests were also run to observe the effect of adding varying amounts of derivative control to the system. This addition was easily made on the analog computer by taking the acceleration, $\dot{V}$, and combining it with the velocity V.

Another test was made to determine the effect of different amounts of constant weight applied to the accelerator pedal. Calibrated weights of 3, 10, and 15 lbs. were hung on the accelerator pedal linkage to simulate a constant foot force on the pedal.

The performance of the system utilizing different size deadbands was observed with a constant weight applied to the accelerator pedal. The width of the deadband was changed by adjusting the reference voltages on the computer comparator relays before each run.

To evaluate the operation of the system at road speeds other than 55 mph, control speeds of 45 and 30 mph were tried. Again a constant weight was applied to the accelerator pedal and the control range was set by adjusting the reference inputs to the computer comparator relays.

Another test was made to determine the best control spring constant. A maximum accelerator pedal load was determined by survey and applied to the pedal. A spring which could handle the maximum load would also be able to handle lighter accelerator pedal loads.

Finally the system response to disturbances such as a varying road slope was tested and recorded. A computer potentiometer was used to feed a disturbance voltage into the automobile transfer function. Both an increasing and decreasing road slope were simulated.

Discussion of Results

Verifying the Existence of Limit Cycles

The results of all experimental tests are listed in Appendix I in graphical form. In accordance with the theoretical analysis the speed control system without derivative control was found to limit cycle as shown in Figure 10. From a time domain viewpoint the instability which caused the system to limit cycle about the deadband was caused by an excessive amount of velocity overshoot and a fast responding control motor. As the velocity left the deadband, either increasing or decreasing, the control motor would completely engage or disengage the control spring before the velocity was able to enter back into the deadband. To reduce the amount of velocity overshoot derivative control was added to the system. This modification caused the deadband relays to monitor a combination of velocity and acceleration $V + K\dot{V}$ instead of velocity V alone.

In testing to see if limit cycles would be tolerable a deadband of 2 mph was utilized and the system response is shown in Figure 11. Whether the limit cycle was tolerable or not was really a question of personal judgement. It was felt that if the accelerator pedal moved noticeably during the limit cycle, then the limit cycle would be annoying and undesirable for long periods of constant operation. Therefore, a plot of accelerator position versus automobile velocity was made. By inspection of Figure 12 it can be seen that the accelerator position moved approximately 3/8 in. back and forth during each limit cycle. This size of movement would definitely be sensed by the driver. Over a period of hours of continuous operation as might be encountered on a long trip, this continuous movement would be undesirable.

Reducing the Control Motor Gain and Increasing the Size of the Deadband

The attempt to eliminate the limit cycles by reducing the control motor gain by a factor of 5 was not successful. Analytically speaking, this would eliminate the limit cycles, but in actual system operation it did not. Any attempts to further reduce the control motor gain would slow the response of the accelerator pedal such that the operation of the system would be sluggish.

A test was run on the system shown in Figure 3 to find out if the limit cycles could be eliminated by increasing the size of the deadband. According to the analytical analysis the limit cycles should not occur for deadbands greater than 4.73 mph. In the experimental test the size of the deadband was increased until the system ceased to limit cycle.

The limit cycling did not cease until the deadband had increased to approximately 18 mph.

Another test was run on the system shown in Figure 3 to determine if the limit cycles could be eliminated by a combination of increasing the size of the deadband and reducing the control motor gain by a factor of 5. The mathematical analysis indicated that for this case the limit cycles should not occur for deadbands greater than 1.2 mph. However, the experimental test showed that the limit cycles were not eliminated until the deadband had increased to approximately 10 mph.

The differences between the analytical calculations and the experimental results were due to inaccuracies in the mathematical model which was used to describe the system in the mathematical analysis. Since the 10 mph and the 18 mph deadband were too large to even consider using, it was decided that the best way to eliminate the limit cycles would be to add derivative control to the system.

Varying the Amount of Derivative Control

The results of adding various amounts of derivative control where K has a value of .75, 1, and 2 are shown in Figures 13, 14, and 15 respectively. The net effect of adding the derivative control was to limit the motor engagement interval which, in turn, caused the system to reach an equilibrium point inside the deadband.

Varying the Amount of Constant Weight on the Accelerator Pedal

The purpose of this test was to show that the system with derivative control would perform as well for a driver with a heavy foot as well as for a driver with a light foot. The results of this test are shown in Figures 16, 17, and 18. The system responded equally well to all weights tested. As expected the velocity overshoot increased as the weight increased. The system response to a constant weight of 5 lbs. on the accelerator pedal was not included because the majority of the other tests were made utilizing this weight. Therefore, to check the system response to this weight reference may be made to Figures 14 or 15.

Varying the Size of the Deadband

The results of using deadbands of 2, 4, and 8 mph with a $.75\dot{V}$ derivative control are shown in Figures 19, 20, and 21. It should be remembered that the graphs are plots of velocity versus acceleration and that the deadband relays monitor a combination of velocity and acceleration. Therefore, when the acceleration is positive, the relays will throw at a velocity, V_t, which is less than the desired throwing velocity, V_D, by an increment equal to .75 times the acceleration at V_t. When the acceleration is negative, the relays will throw at a velocity, V_t, which is greater than the desired throwing velocity, V_D, by an increment equal to .75 times the acceleration at V_t. This operation causes no real problem because as the automobile approaches a steady state velocity, the absolute value of the acceleration becomes less and less. So in effect, the variable deadband adjusts itself to the desired throwing velocities as the acceleration approaches zero.

As the deadband increases in size the accuracy for settling at the desired cruising velocity is decreased as shown by comparing Figures 20 and 21. It is evident that the probability of settling at the desired cruising speed of 55 mph is much greater with a deadband of 4 mph than with a deadband of 8 mph. Another disadvantage of a large deadband is that it would take longer to detect a changing velocity caused by a change in road slope such as a hill.

The small deadband of 2 mph showed a tendency to try to limit cycle. However, this undesirable tendency can easily be eliminated by increasing the amount of derivative control.

Varying the Controlled Cruising Speed

To evaluate the system performance at controlled road speeds other than 55 mph tests were run for speeds of 30 and 45 mph. The results of these tests are shown in Figure 22. The control system performed very well at both of these speeds.

Varying the Control Spring Constant

The purpose of varying the control spring constant was to determine which springs would be strong enough for all driver situations. The most direct way to accomplish this was to find an approximate maximum load for the accelerator pedal and use it to determine which springs were strong enough for the system to operate properly. If a spring can operate properly under a maximum load, then it follows that it will operate properly for smaller loads.

To arrive at a maximum load on the accelerator pedal a survey utilizing six men was conducted. First, data of weight hung from the accelerator pedal linkage versus linear potentiometer voltage was recorded as shown in Figure 23. The linear potentiometer voltage was used to indicate the position of the throttle linkage. Then the six men were asked to lay their foot in a relaxed position on the accelerator pedal to simulate highway driving conditions. The linear potentiometer voltage was recorded and the results of this survey were plotted on the calibrated curve of Figure 23 to determine the equivalent weight of each individual foot.

From the results of Figure 23 a weight of 20 lbs., which is greater than the weight placed on the pedal by any of the six men, was chosen to be hung from the accelerator pedal linkage.

The results of testing control springs with constants of 9, 7, and 5 lbs/in. under maximum loading conditions are shown in Figures 24, 25, and 26. The 9 lb./in. and 7 lb./in. control springs were able to operate sufficiently with a velocity overshoot of 7 mph. Although this 7 mph overshoot is rather large, it would not be objectionable. In the case of the 5 lbs./in. control spring the system obtained the desired cruising speed but the velocity overshoot was 16 mph, which would be objectionable.

It should be noted that the control springs used in all of these tests were positioned as shown in Appendix II (Figure 31) at an angle of 44.8 degrees with the horizontal plane. During the test under maximum loading conditions the 7 lbs./in. control spring was stretched 1.75 inches which produced a resultant force of 12.3 lbs at an angle of 44.8 degrees with the horizontal. Therefore, for the system to operate properly under all conditions, any control spring, which is used and positioned at any arbitrary angle with the horizontal, must be sufficiently strong to produce a resultant force which is equivalent to the 12.3 lbs. applied at 44.8 degrees.

Effect of Disturbances

The system response to a disturbance such as a decreasing road slope, which would cause an increase in velocity and acceleration, is shown in Figure 27. The velocity was initially constant at 55.2 mph as shown by point 1 on Figure 27. When the disturbance voltage was fed to the automobile transfer function, the velocity increased until the upper velocity limit of the deadband was reached at point 2. At this instant the control motor was engaged to tighten the control spring until the velocity dropped below the upper deadband limit at point 3. At point 3 the control motor was turned off and the velocity steadily decreased to point 4 where the control motor was engaged to steadily loosen the control spring. At point 5 the control motor was disengaged and the velocity increased until it reached a constant 54.8 mph.

The system also responded well to an increasing road slope disturbance as shown in Figure 28. The velocity was initially constant at 55 mph as shown by point 1. When the disturbance voltage was fed to the automobile transfer function, the velocity decreased until the lower velocity limit of the deadband was reached at point 2. Then by engaging and disengaging the control motor the system eventually returned to a constant velocity at point 8. The reason the accelerations in Figures 27 and 28 did not end exactly at zero was probably due to a small drift in the analog computer.

CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

Conclusions

The automatic speed control system without compensation was found to be stable but tended to limit cycle about the velocity deadband. This meant that the automobile velocity would oscillate about the control range and never settle at a constant value. The fact that the accelerator pedal moved so noticeably during the limit cycling discouraged the idea that it might be possible to tolerate the limit cycles.

The attempts to eliminate the limit cycles by decreasing the control motor gain were unsuccessful. The limit cycles were eliminated by increasing the size of the deadband to 18 mph. However, a deadband this large would be impractical for actual system operation.

Compensating the system by adding derivative control completely eliminated the limit cycling. Since there was no way to obtain $\dot{V}$ from the automobile on the originally designed system, additional system components would be needed to obtain derivative control. It was decided that derivative control on the automobile could be obtained by using a small tachometer generator and a derivative resistor-capacitor network to generate V and $\dot{V}$ respectively. The results of using the generator and derivative network were not obtained because installation of the control system on an automobile was beyond the scope of this thesis.

The system performed equally well: 1) for both a heavy and a light load on the accelerator pedal, and 2) at controlled cruising speeds from 30 to 55 mph.

Although the system worked well for various size deadbands, the chances of settling at the desired cruising velocity were much better with a small deadband.

A control spring with a constant less than 7 lbs./in. would not be sufficiently strong to operate the system with a maximum load on the accelerator pedal.

The system response to an external disturbance such as an increasing road slope was very good.

One of the great advantages of this speed control system is the safety which it offers. At any time regardless of the position of the control spring, when foot pressure is removed from the accelerator pedal, the stationary spring will return the accelerator pedal to its off position. This means that for emergency stops an automobile would react the same regardless of whether the control system was on or off.

Another advantage of this system is that it would be more economical to manufacture than any of the popular speed control systems which are being used today.

Recommendations

From an economical view point the best approach for future study would be to develop a way to compensate the system without using a generator and derivative network. If a way could be found to limit the control motor engagement time by some type of capacitor-resistor network, this would probably eliminate the limit cycles.

The next logical area to explore in the development of this speed controller would be to actually road test the system. A speed controller could be built on the information given in this thesis and tested on a motor vehicle. This would furnish conclusive proof of the system's effectiveness.

APPENDIX I

GRAPHICAL RESULTS OF THE EXPERIMENTAL INVESTIGATIONS

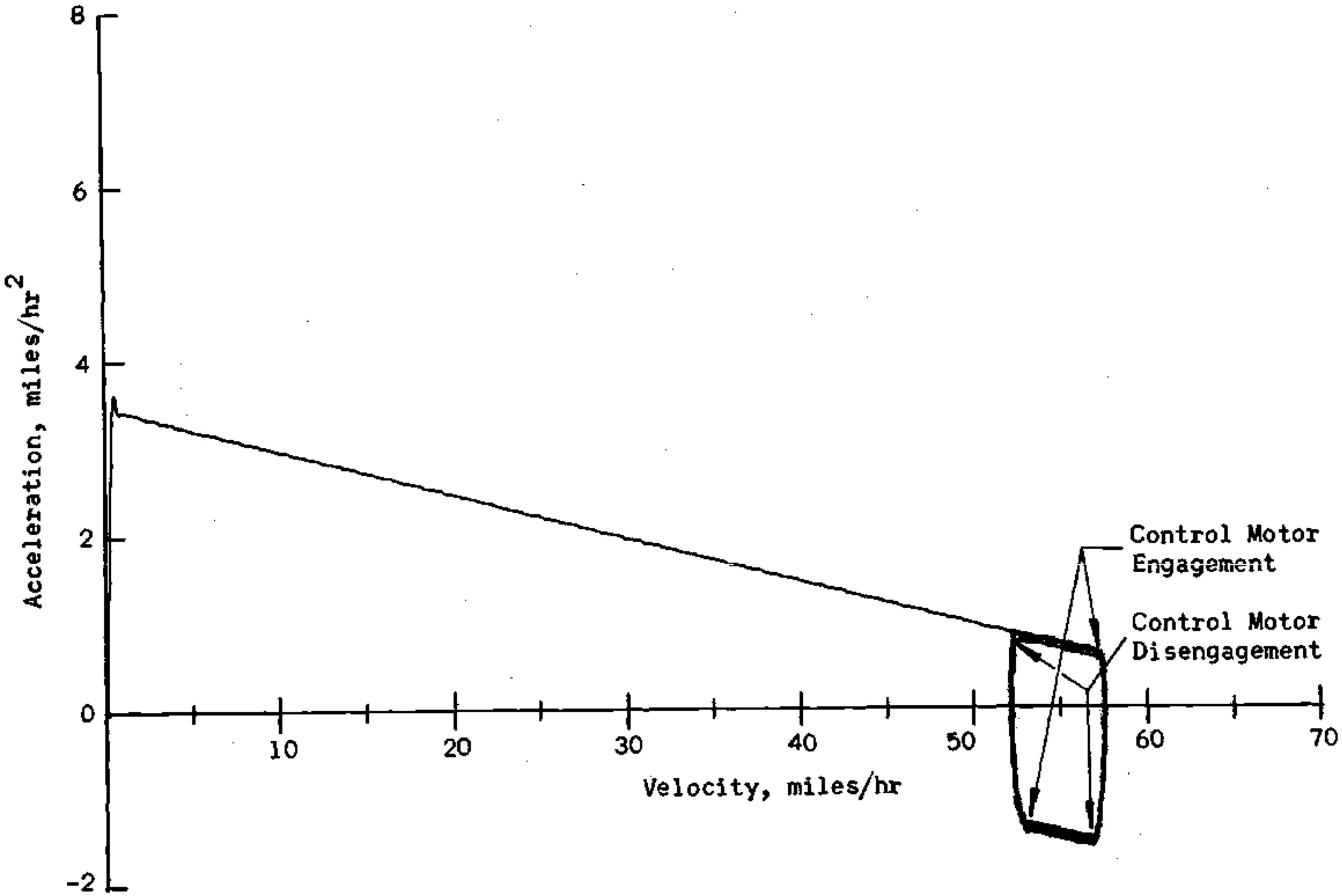

Figure 10. Automobile Response with No Derivative Control.

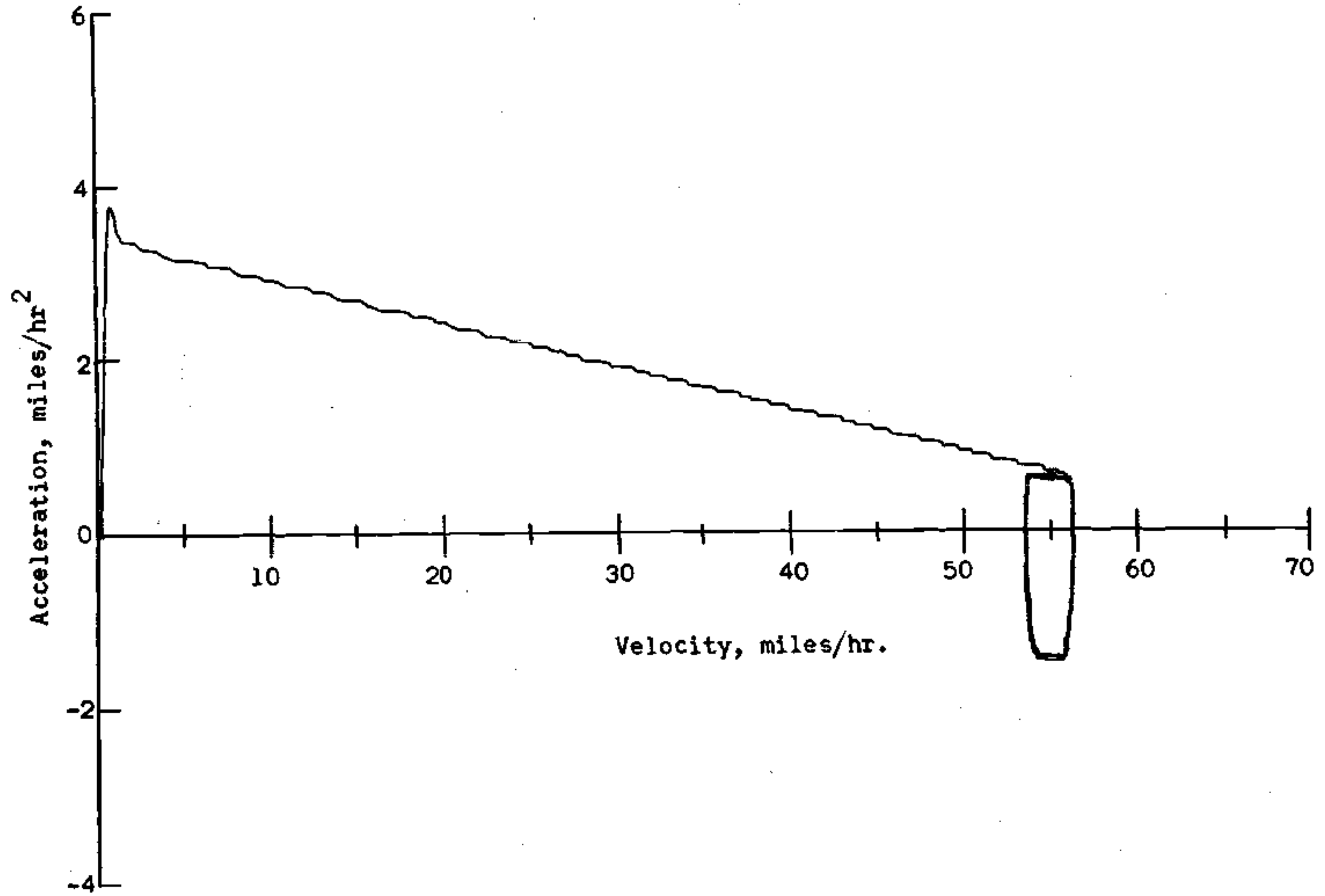

Figure 11. Automobile Response with No Derivative Control and a Deadband of 2 mph.

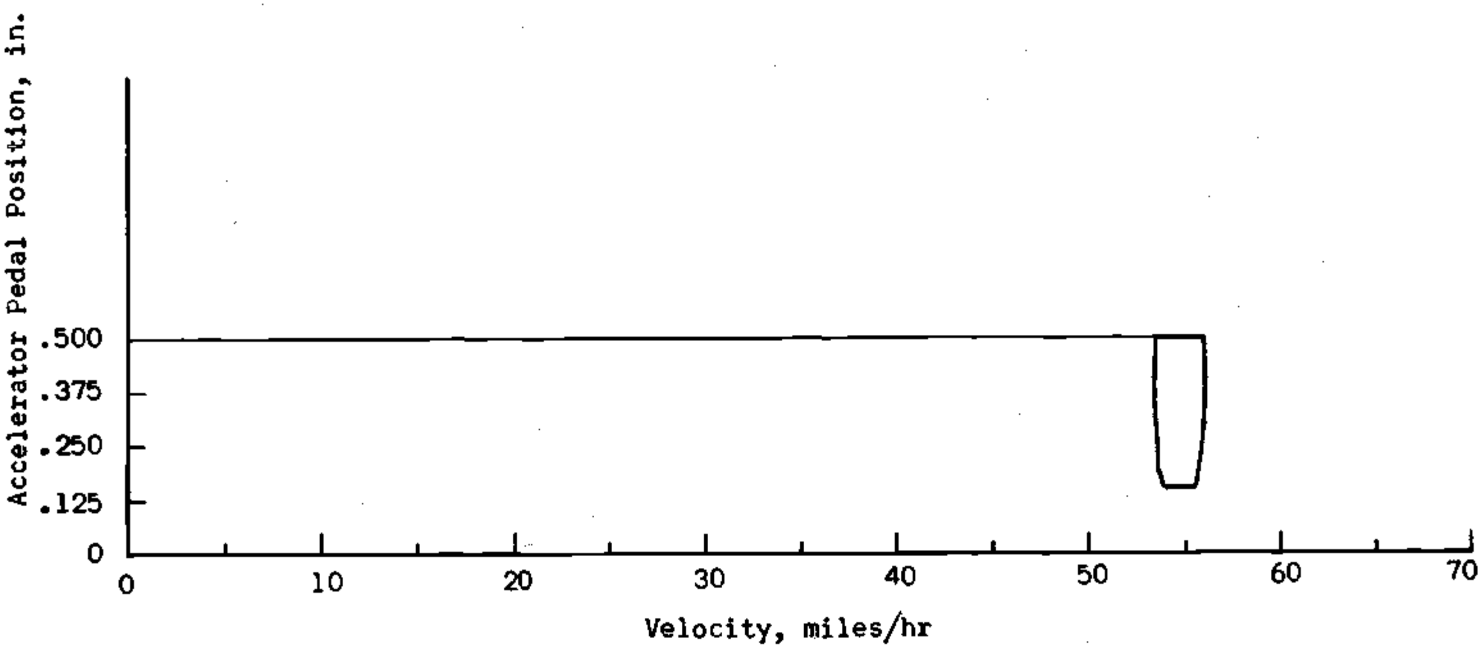

Figure 12. Accelerator Pedal Position for a Step Input with No Derivative Control and a Deadband of 2 mph.

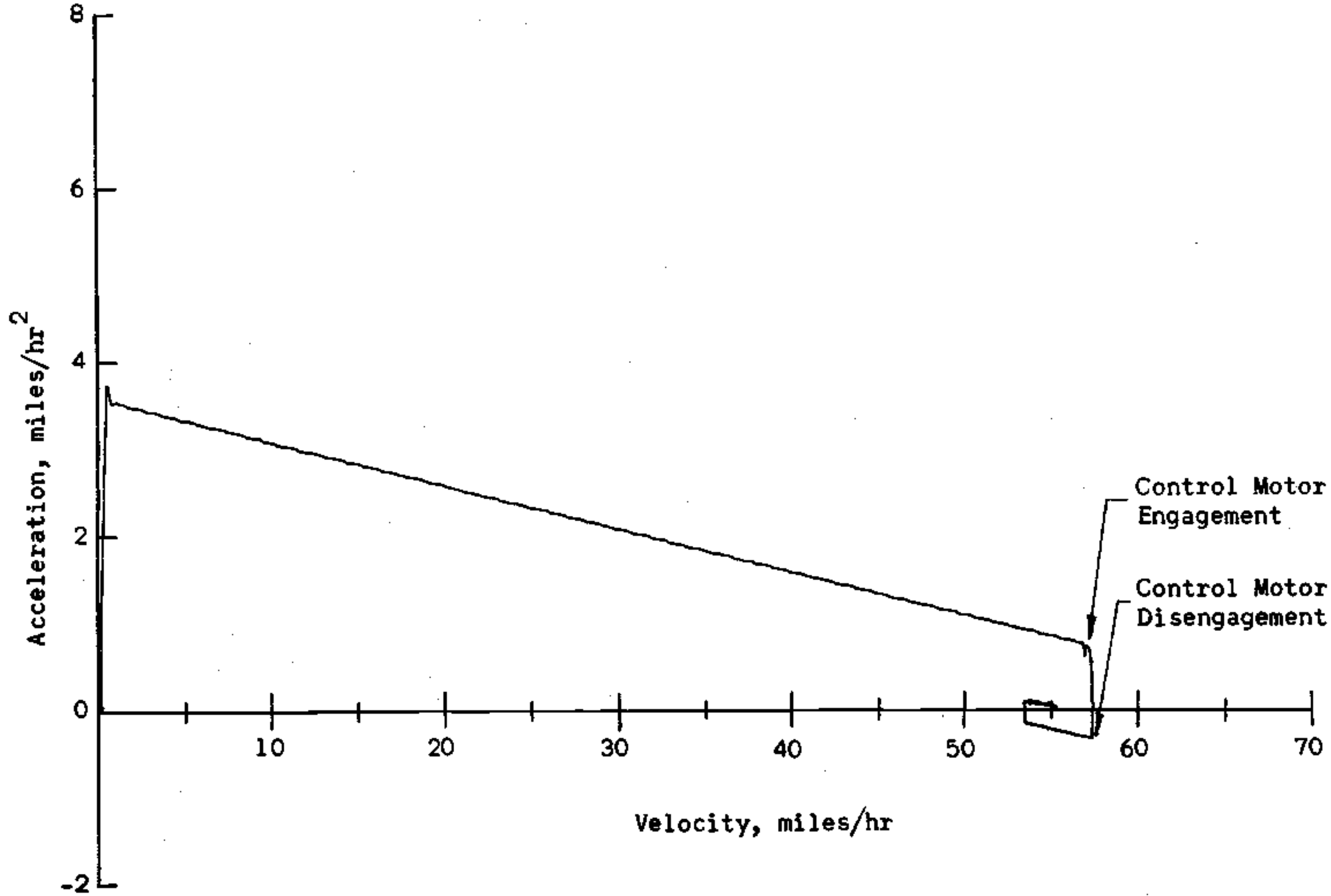

Figure 13. Automobile Response with .75 $\dot{V}$ Derivative Control.

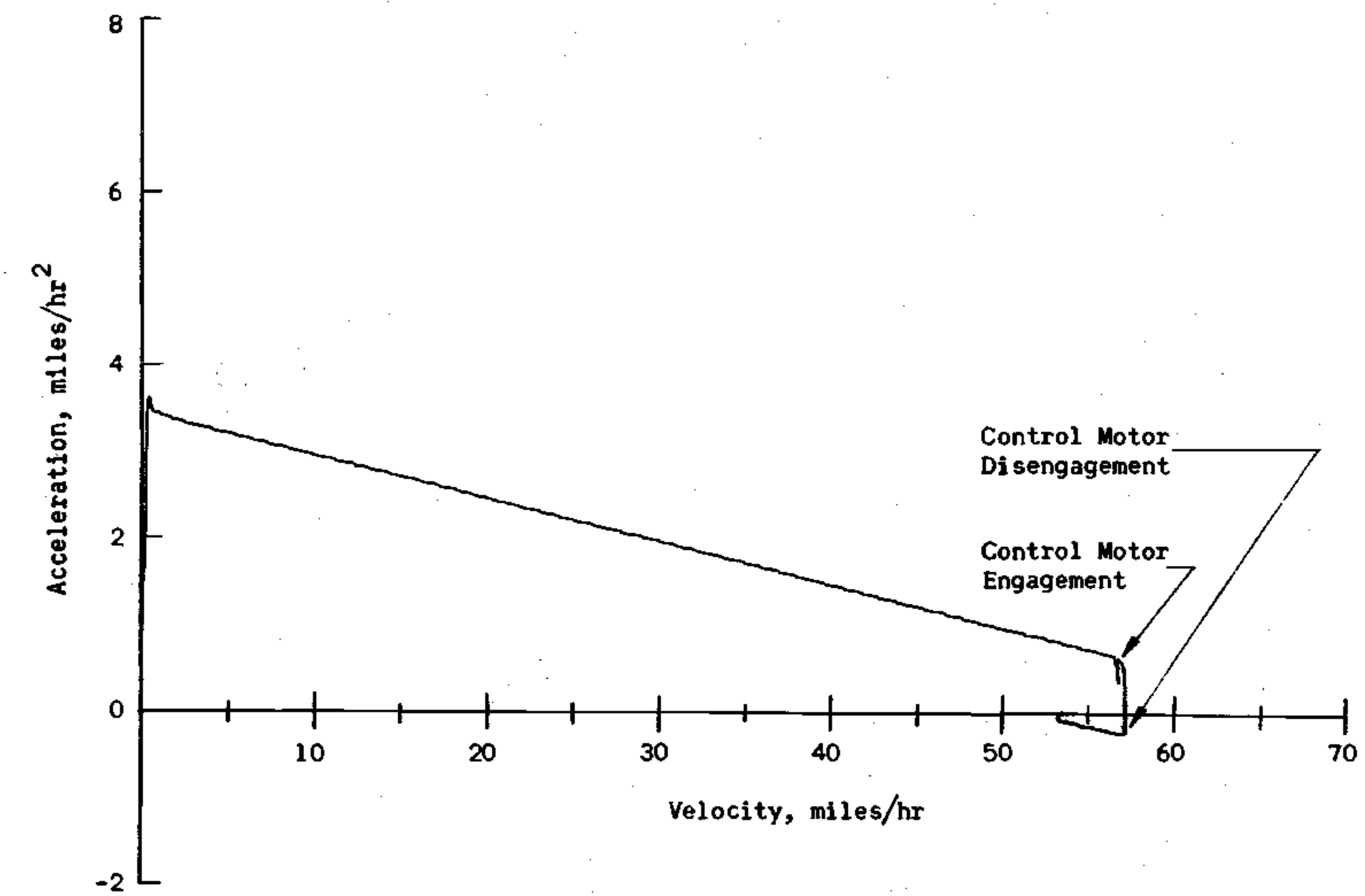

Figure 14. Automobile Response with $1\dot{V}$ Derivative Control.

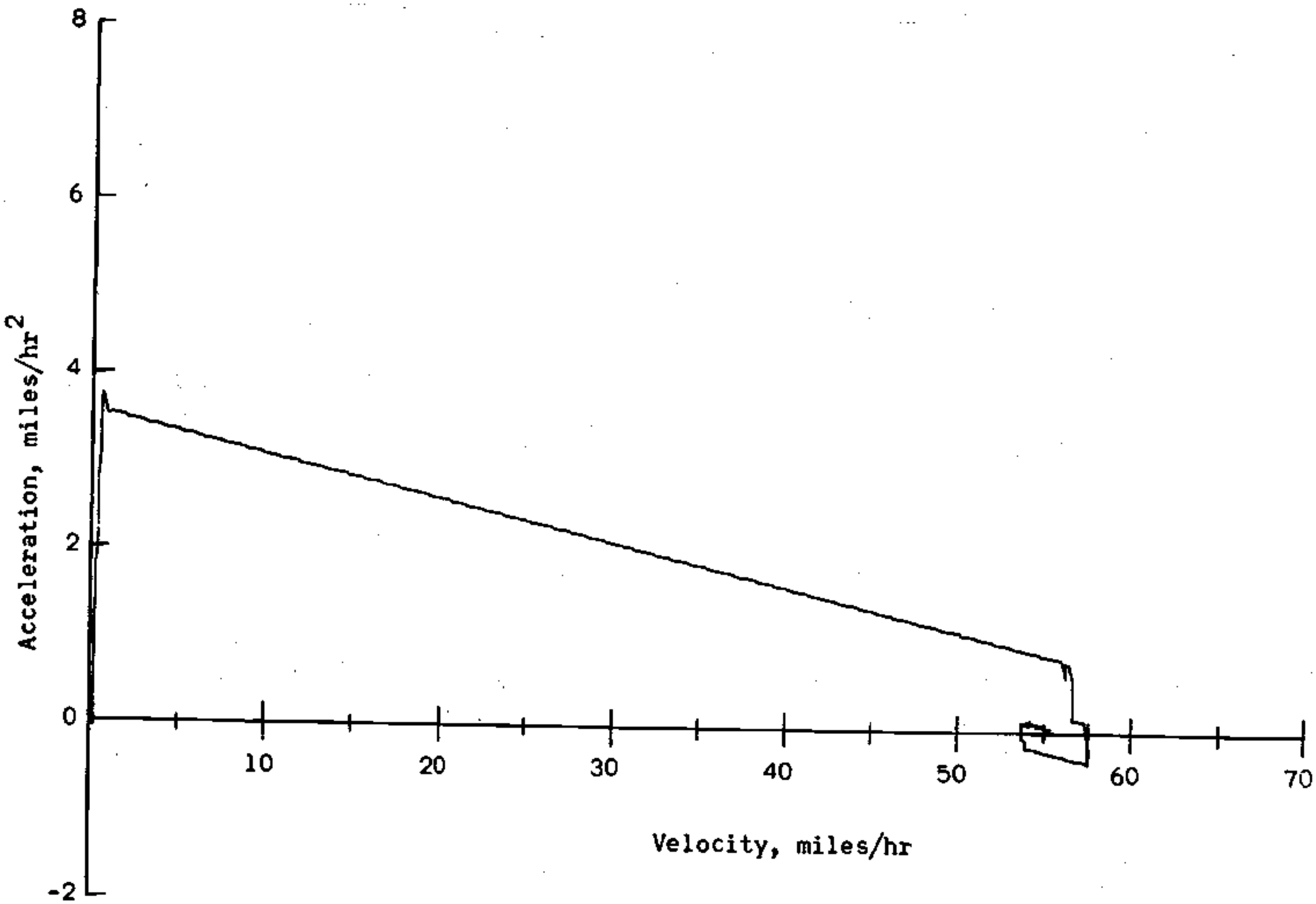

Figure 15. Automobile Response with 2 $\dot{V}$ Derivative Control.

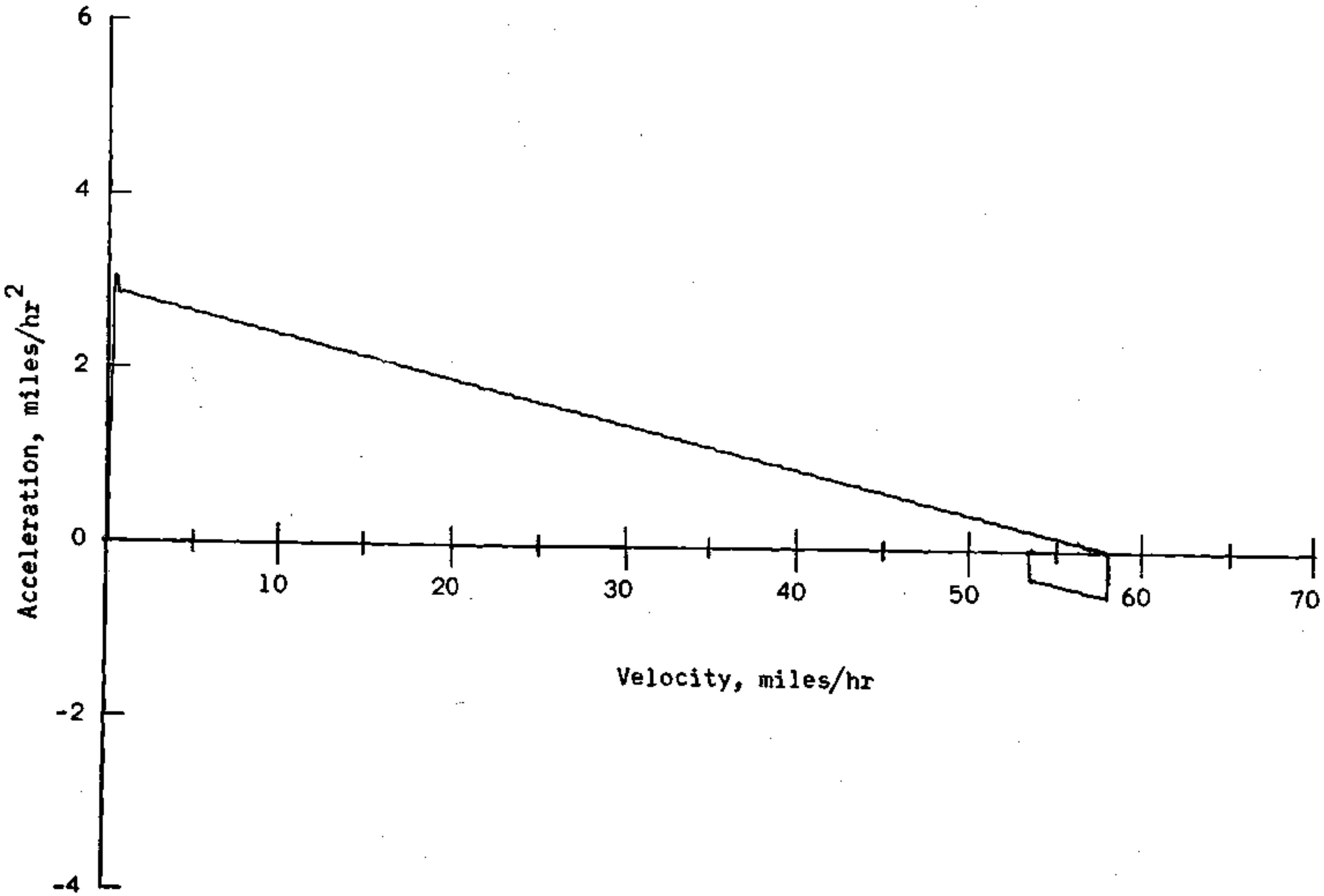

Figure 16. Automobile Response with a Constant Weight of 3 lbs. on the Accelerator Pedal.

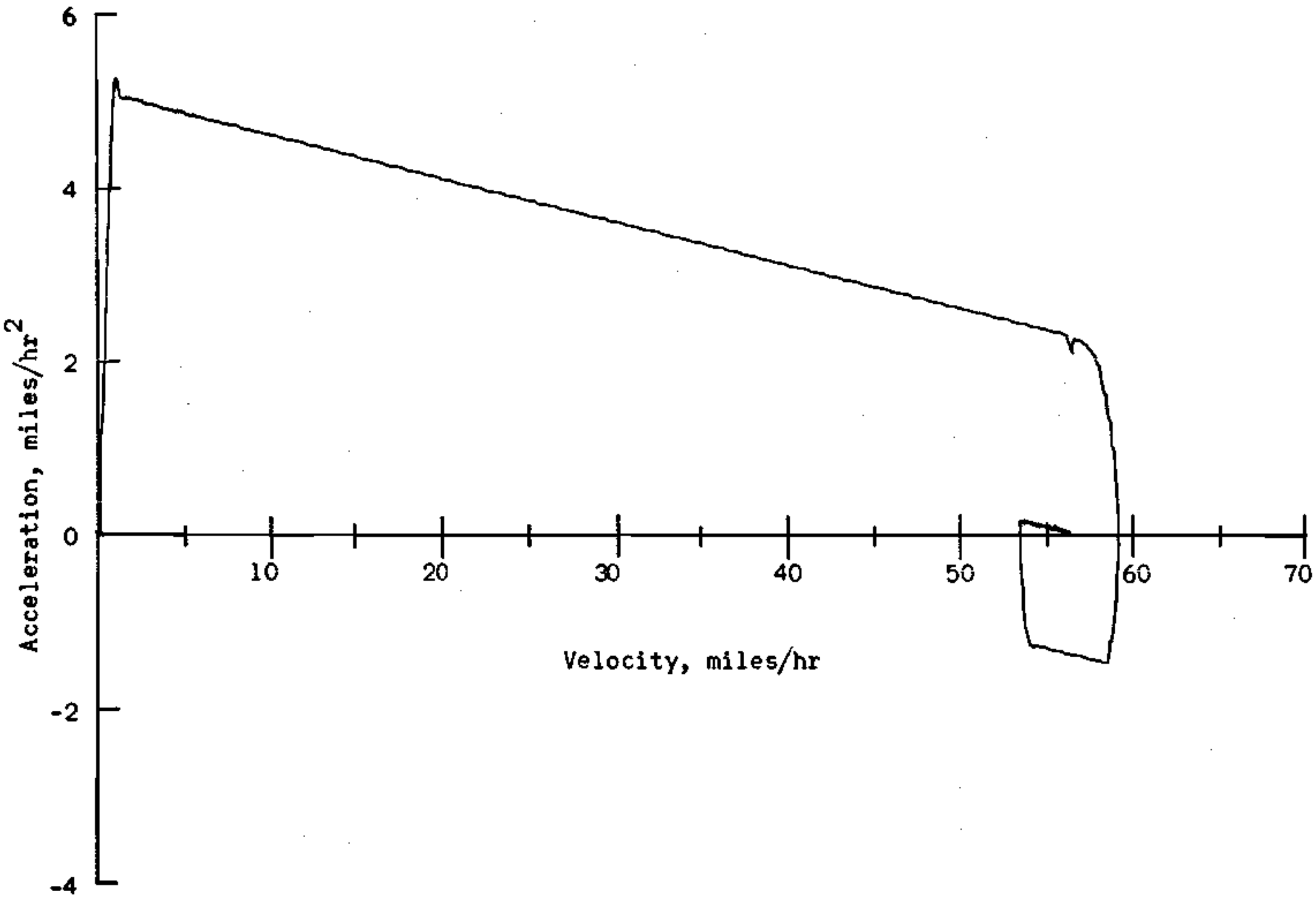

Figure 17. Automobile Response with a Constant Weight of 10 lbs. on the Accelerator Pedal.

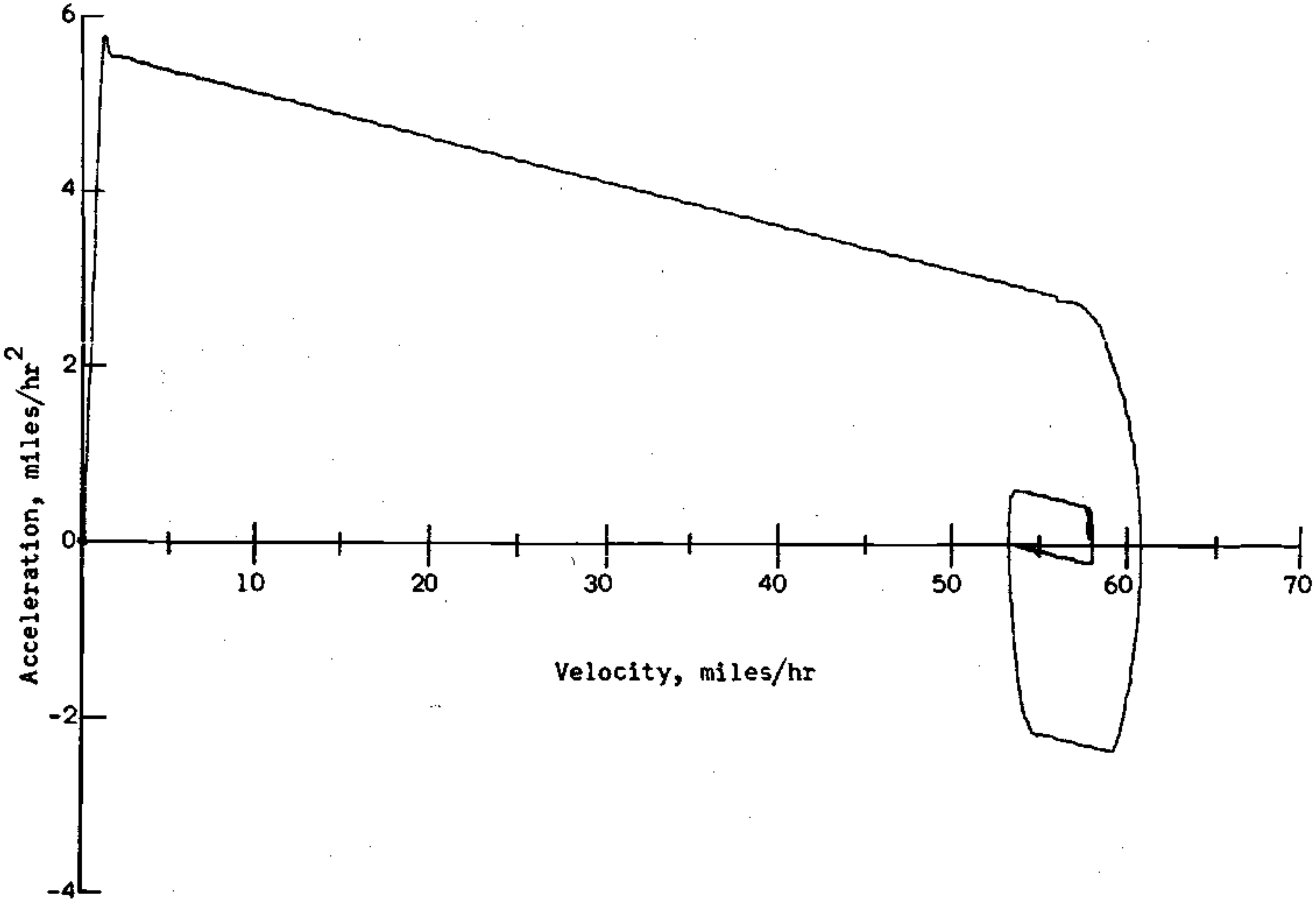

Figure 18. Automobile Response with a Constant Weight of 15 lbs. on the Accelerator Pedal.

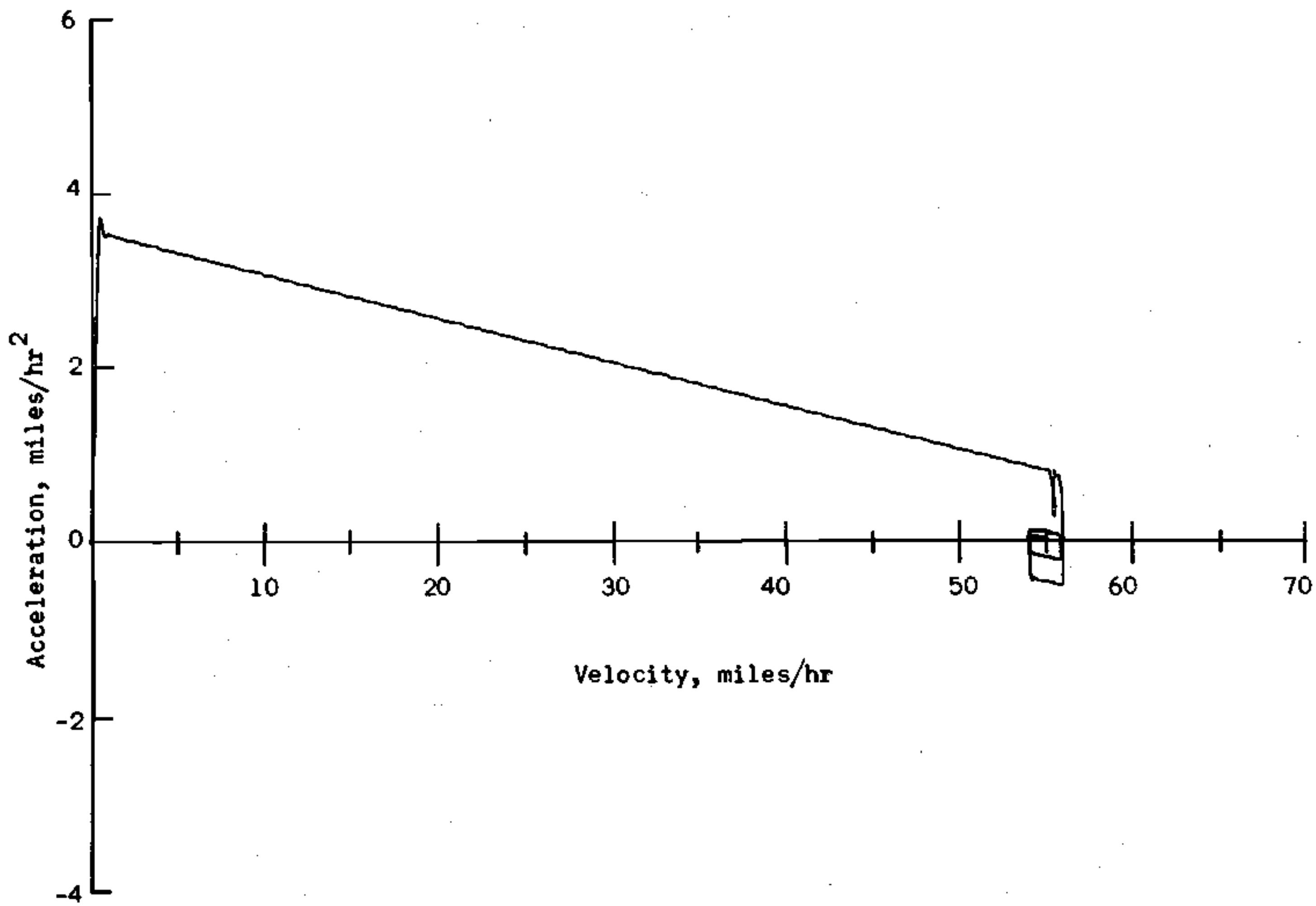

Figure 19. Automobile Response with a Deadband of 2 mph.

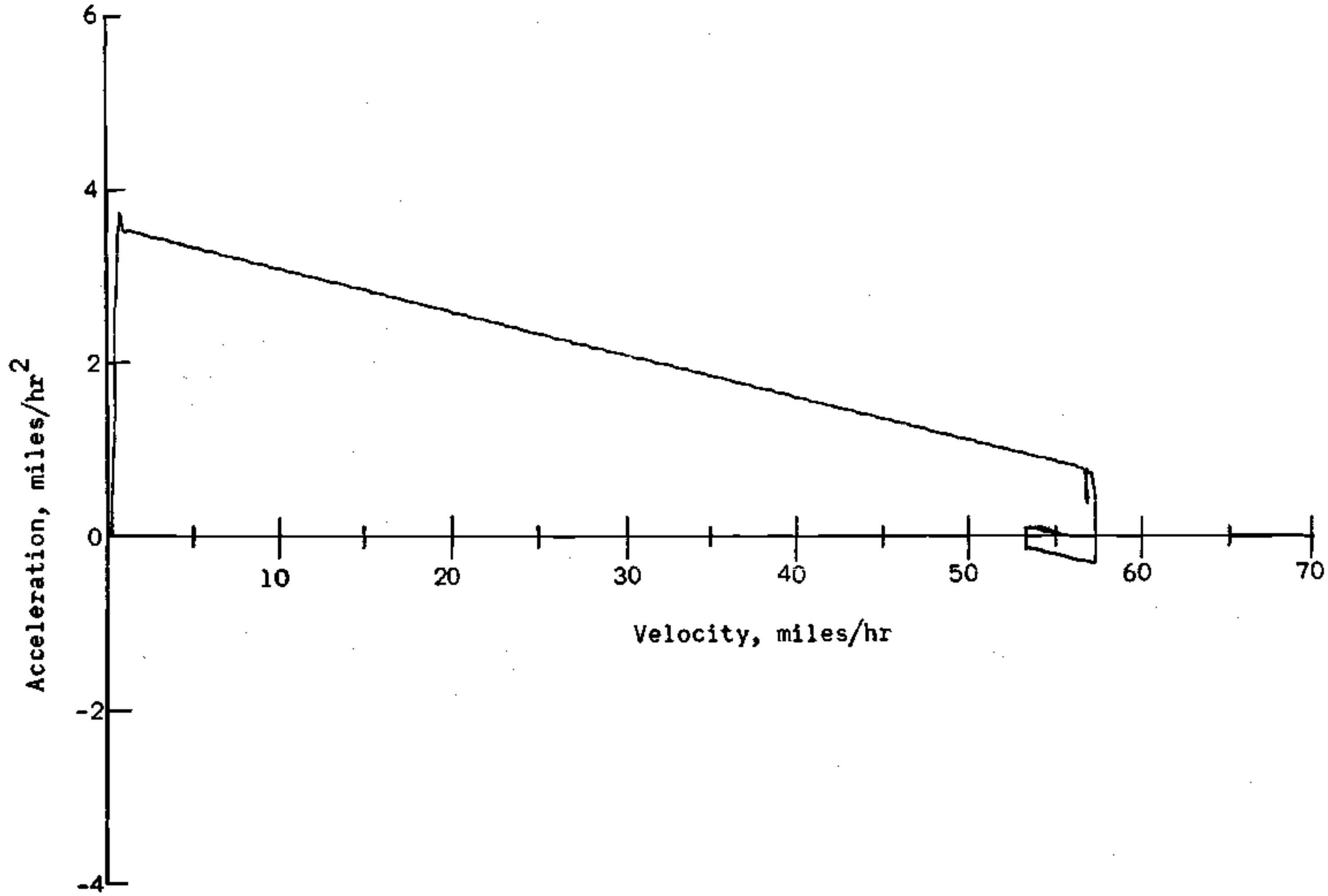

Figure 20. Automobile Response with a Deadband of 4 mph.

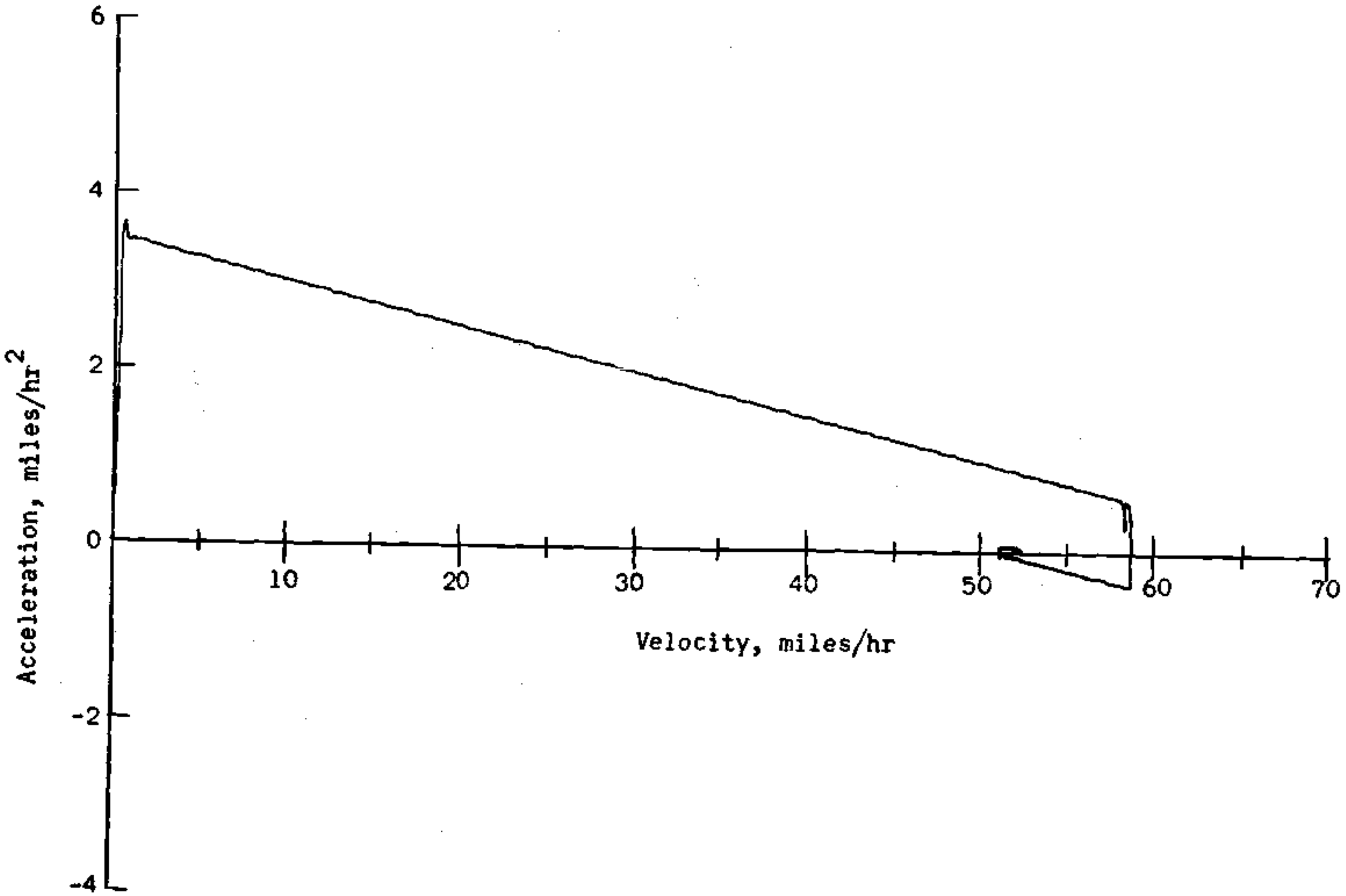

Figure 21. Automobile Response with a Deadband of 8 mph.

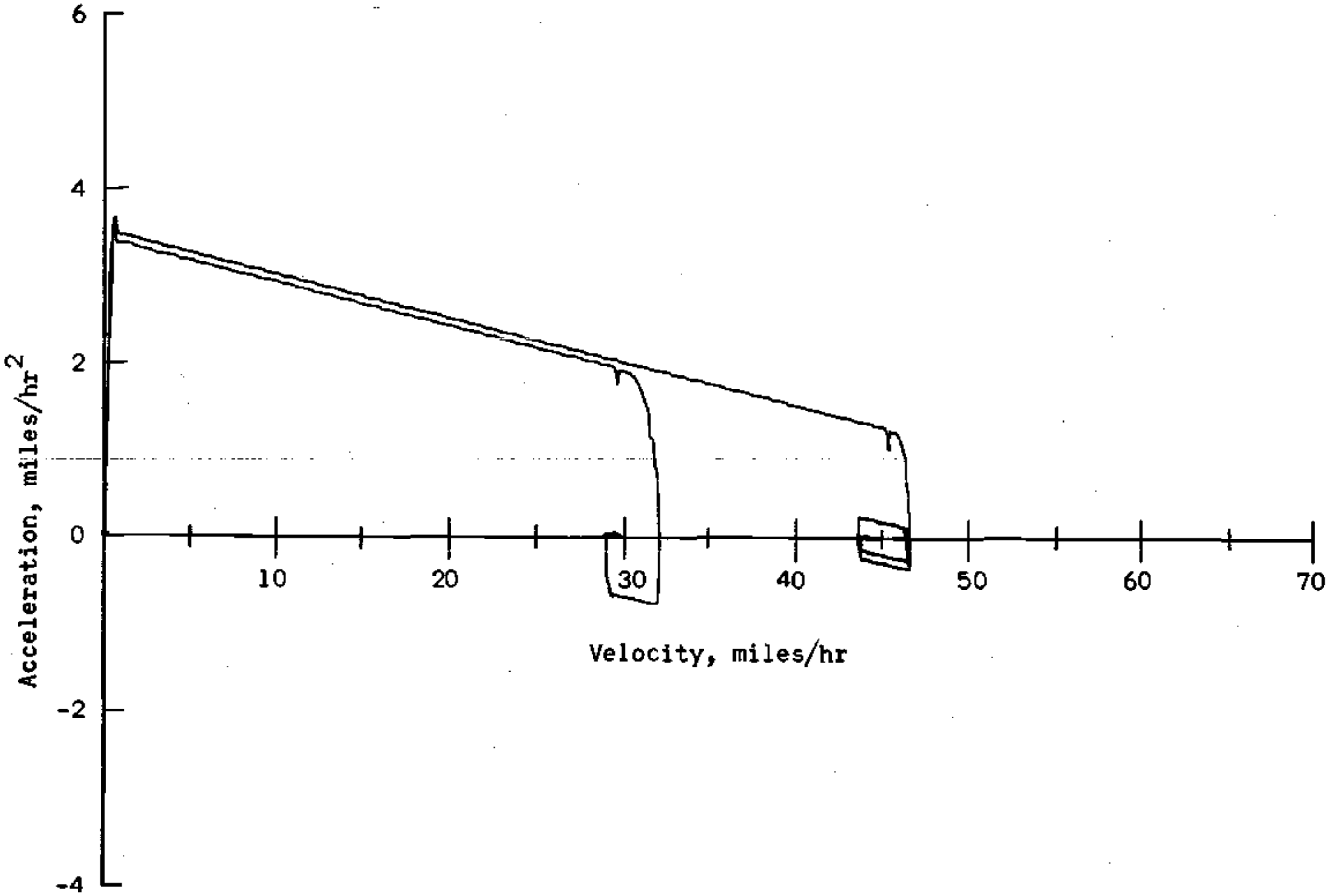

Figure 22. Automobile Response for Controlled Cruising Speeds of 30 and 45 mph.

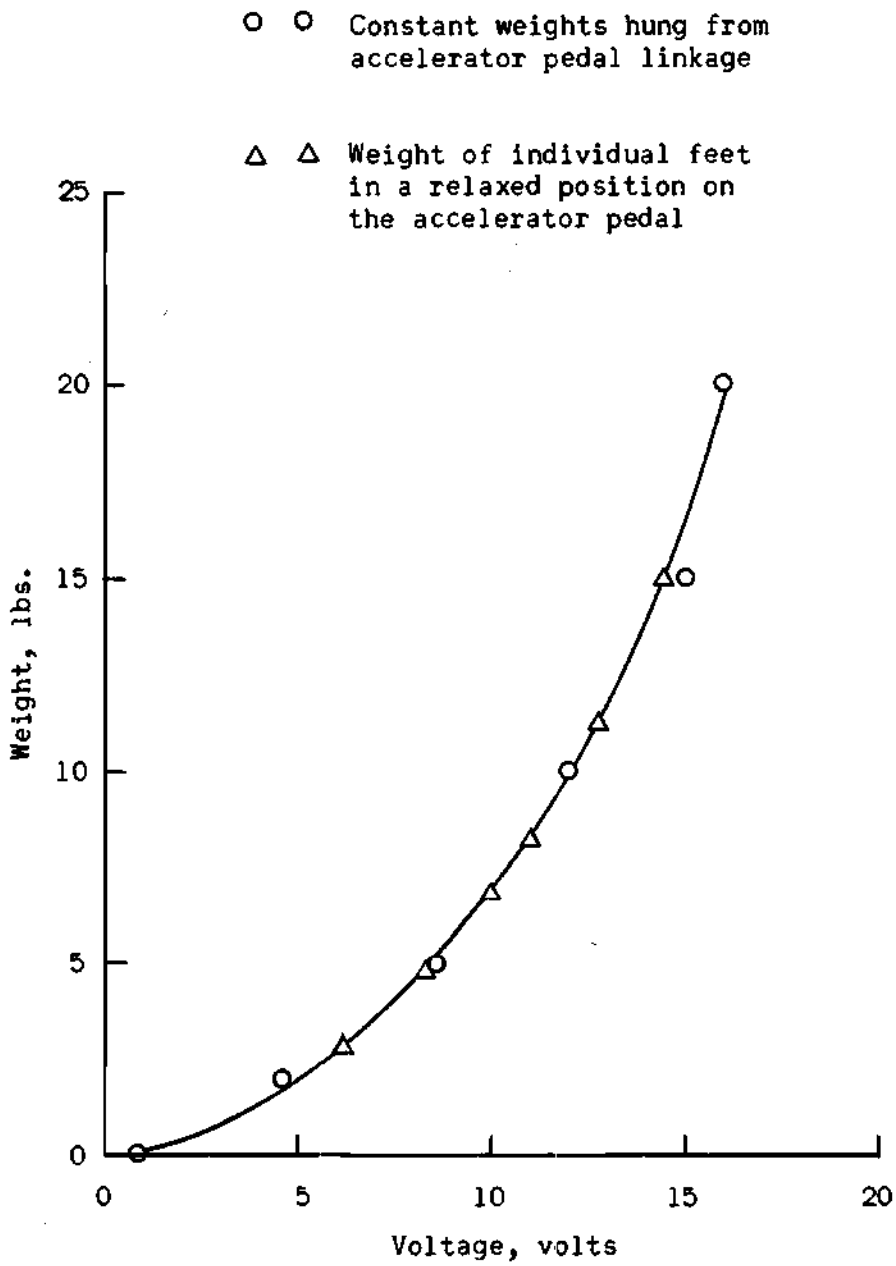

Figure 23. Comparison of Weights Applied to the Accelerator Pedal.

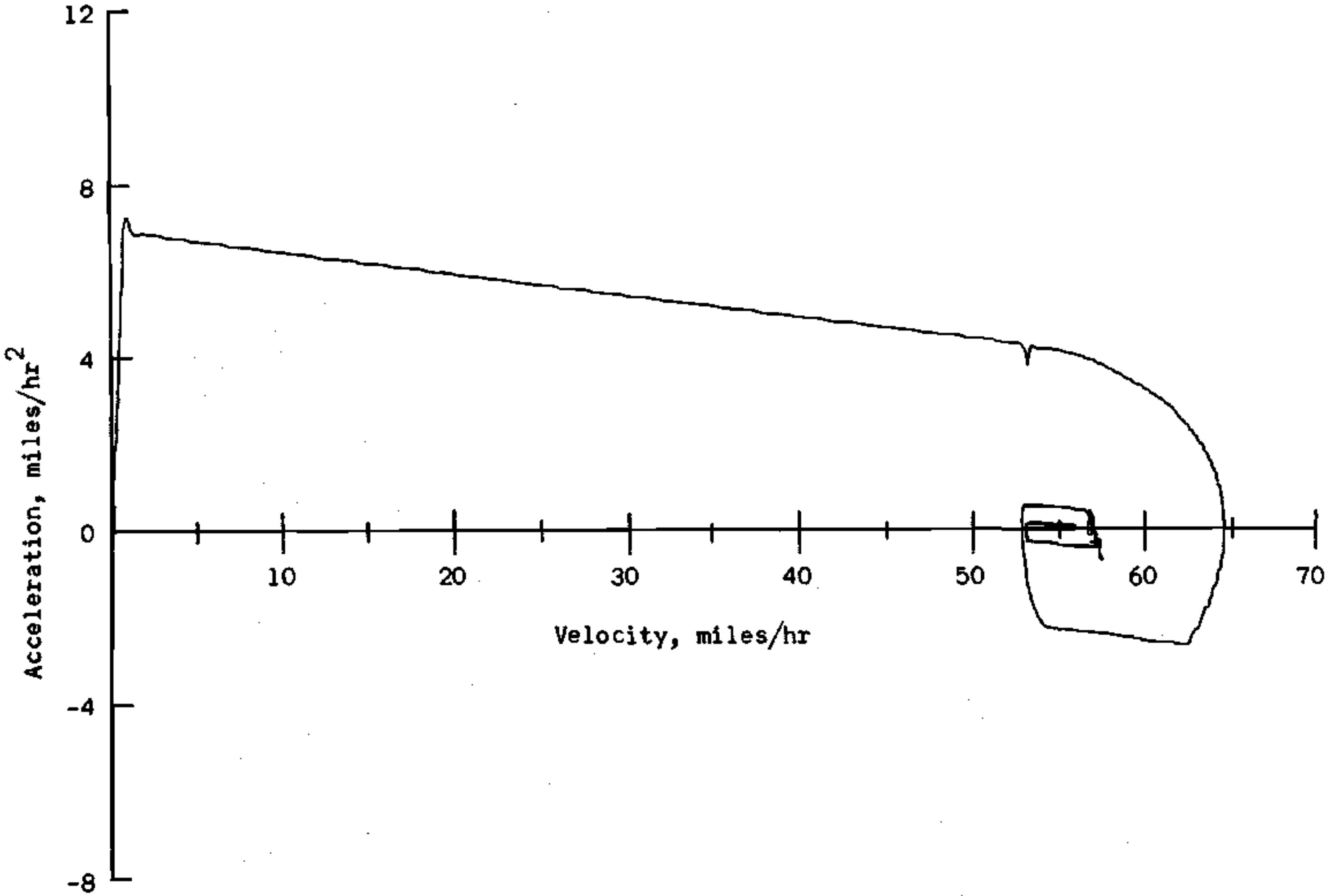

Figure 24. Automobile Response with a Maximum Load on The Accelerator Pedal and Control Spring with a 9 lb/in. Constant.

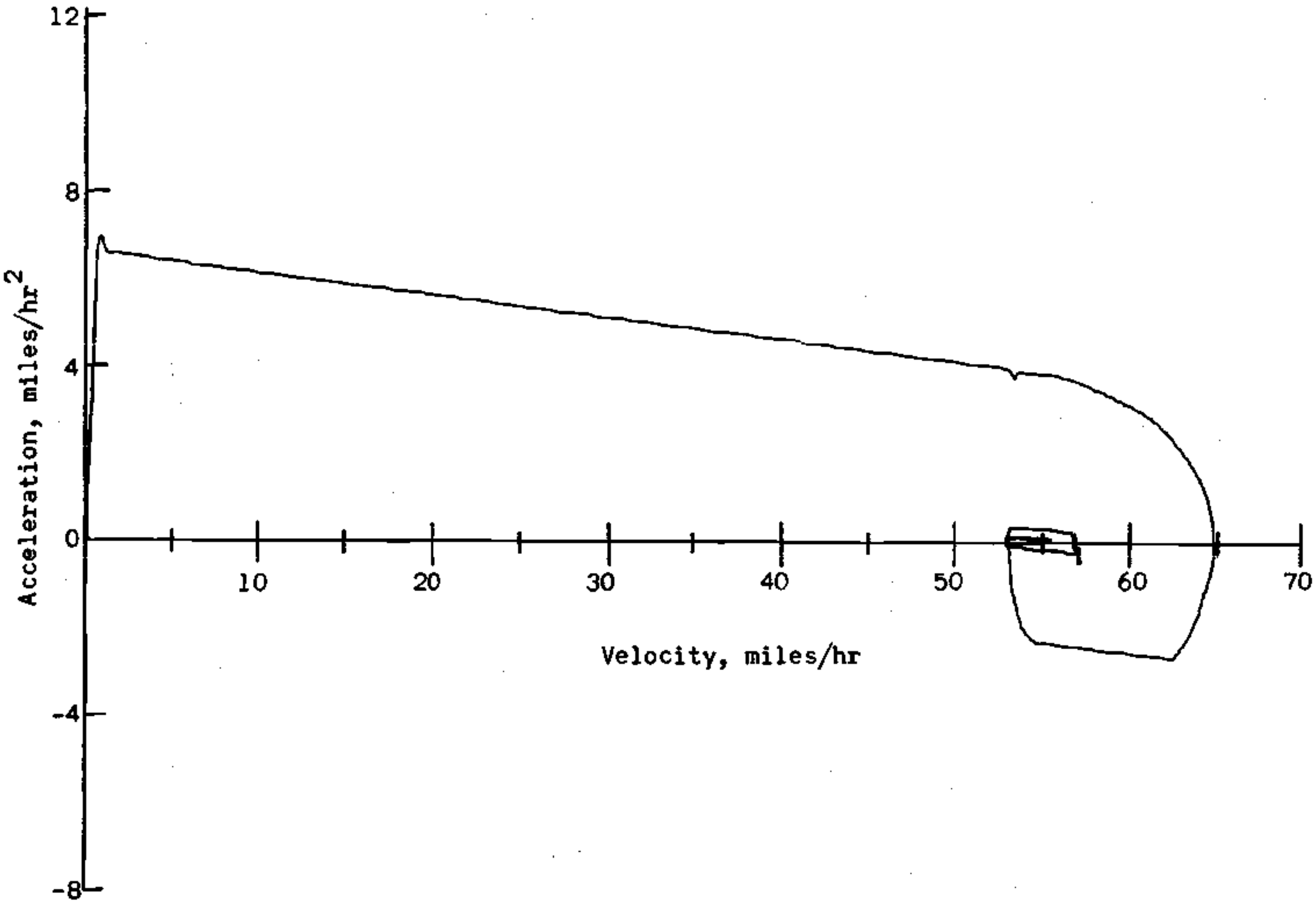

Figure 25. Automobile Response with a Maximum Load on the Accelerator Pedal and Control Spring with a 7 lb/in. Constant.

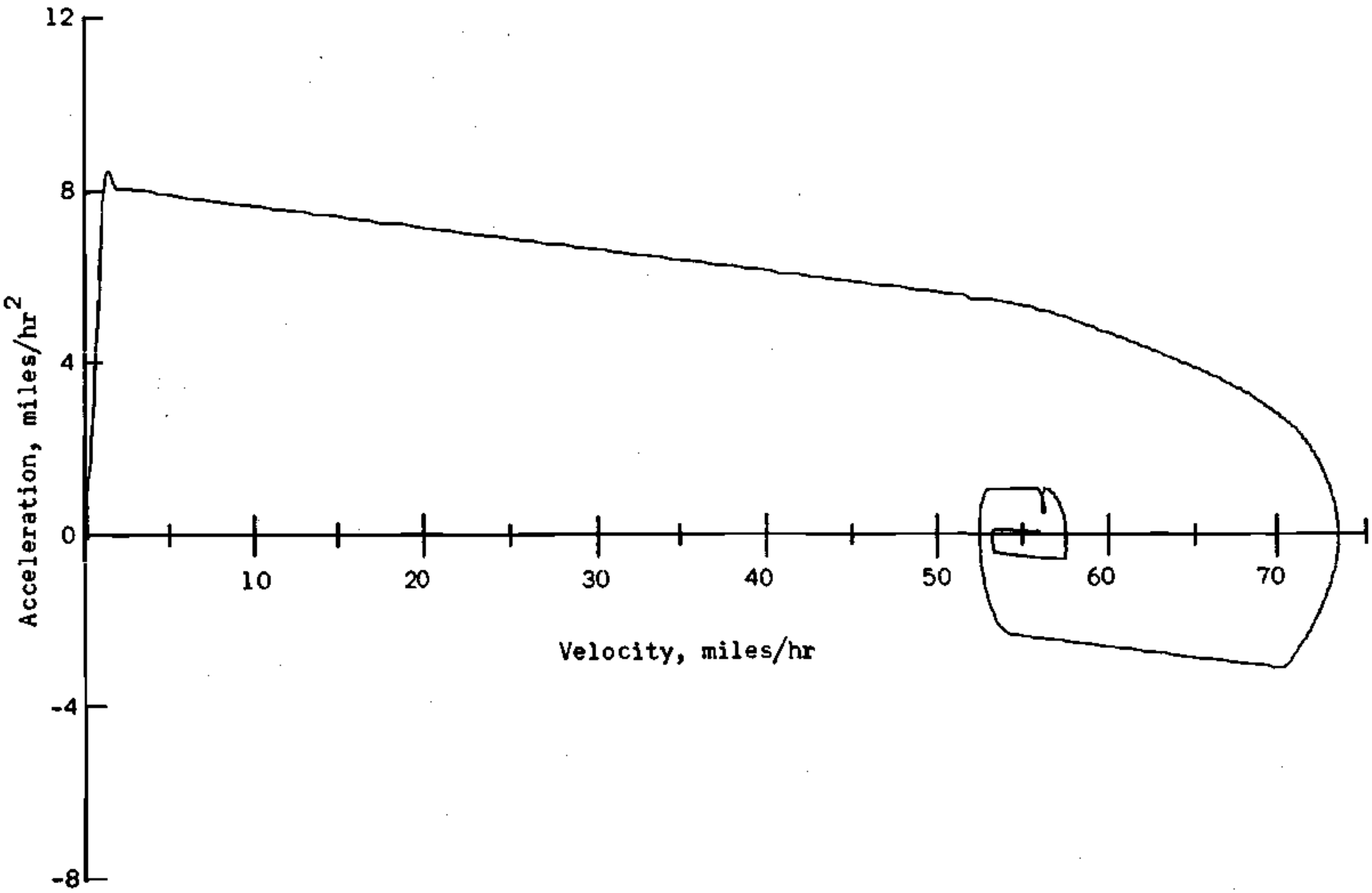

Figure 26. Automobile Response with a Maximum Load on the Accelerator Pedal and Control Spring with a 5 lb/in. Constant.

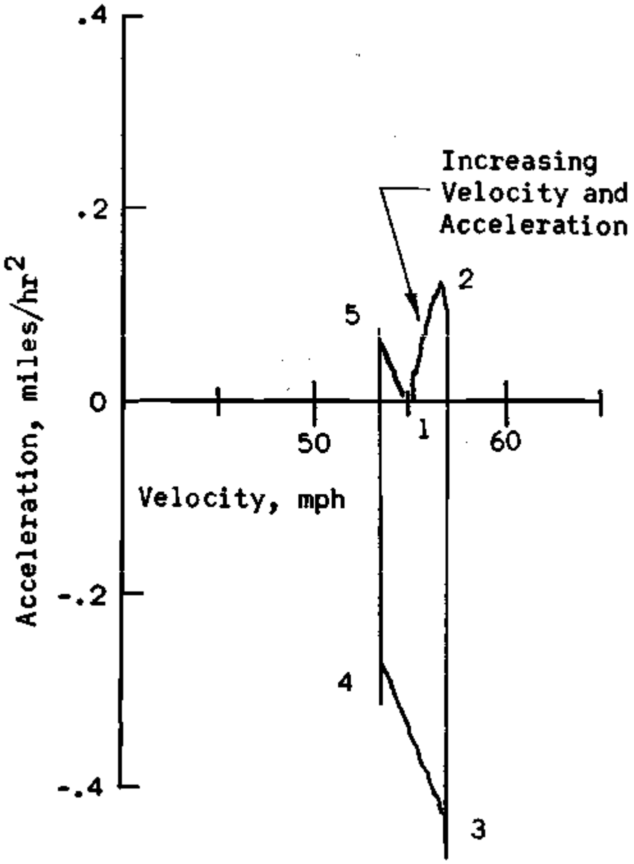

Figure 27. Automobile Response to a Decreasing Road Slope Disturbance.

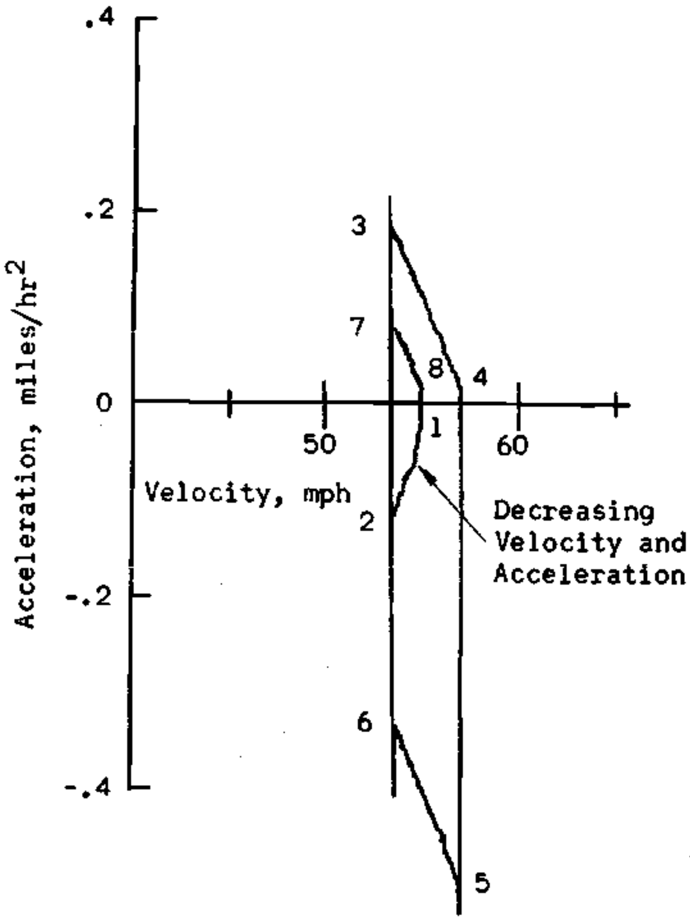

Figure 28. Automobile Response to an Increasing Road Slope Disturbance.

APPENDIX II

EXPERIMENTAL EQUIPMENT

Automobile Response Measuring Apparatus

The apparatus for taking velocity response data on the Oldsmobile 88 is shown in Figure 29. A steel extension rod clamped to the throttle linkage was used to manually apply the step inputs. A metal scale attached to the dashboard was used to measure the movement of the extension rod. A metal clamp served as an adjustable stop to limit the magnitude of the step input. The relationship between carburetor arm linkage movement and extension rod movement is shown in Figure 30. Six Fischer Scientific Company stop watches were used to record time on each test run. The watches were started together and then stopped individually at increments of 2 1/2, 5, 7 1/2, 10, 12 1/2, 15 mph. from the starting velocity.

Laboratory Experimental Apparatus

A diagram of the laboratory experimental apparatus is shown in Figure 31. The accelerator pedal and linkage arm were standard Oldsmobile parts. The stationary and control springs were wound on a lathe and made from music wire. A linear Markite potentiometer No. 4709 with a resistance of 20,000 ohms/inch was used to transmit the linkage position to the analog computer.

In order to simulate the linkage properly, the drag effect of the carburetor and the automatic transmission passing gear linkage had to be included in the test set up. To measure this effect the carburetor

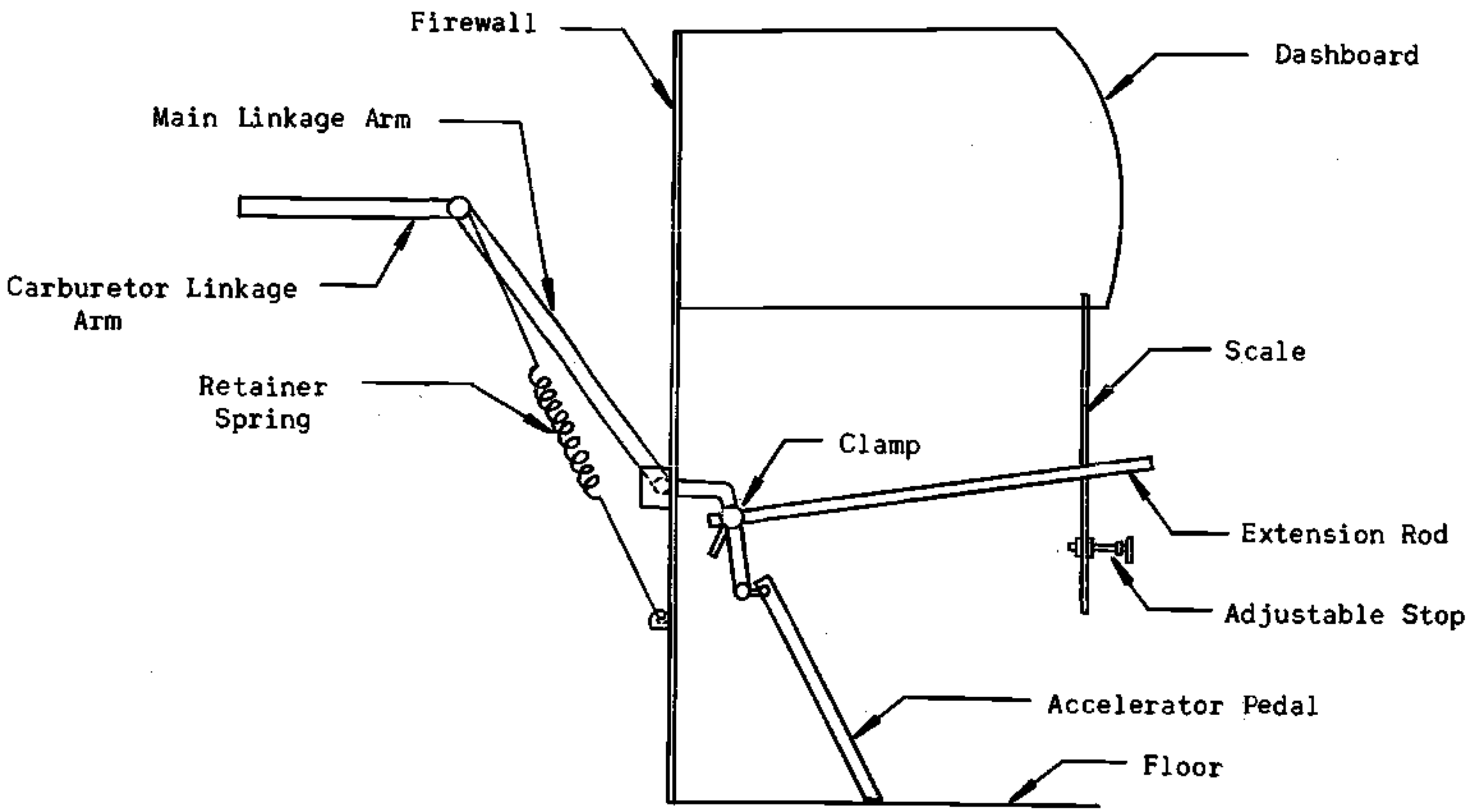

Figure 29. Automobile Response Measuring Apparatus.

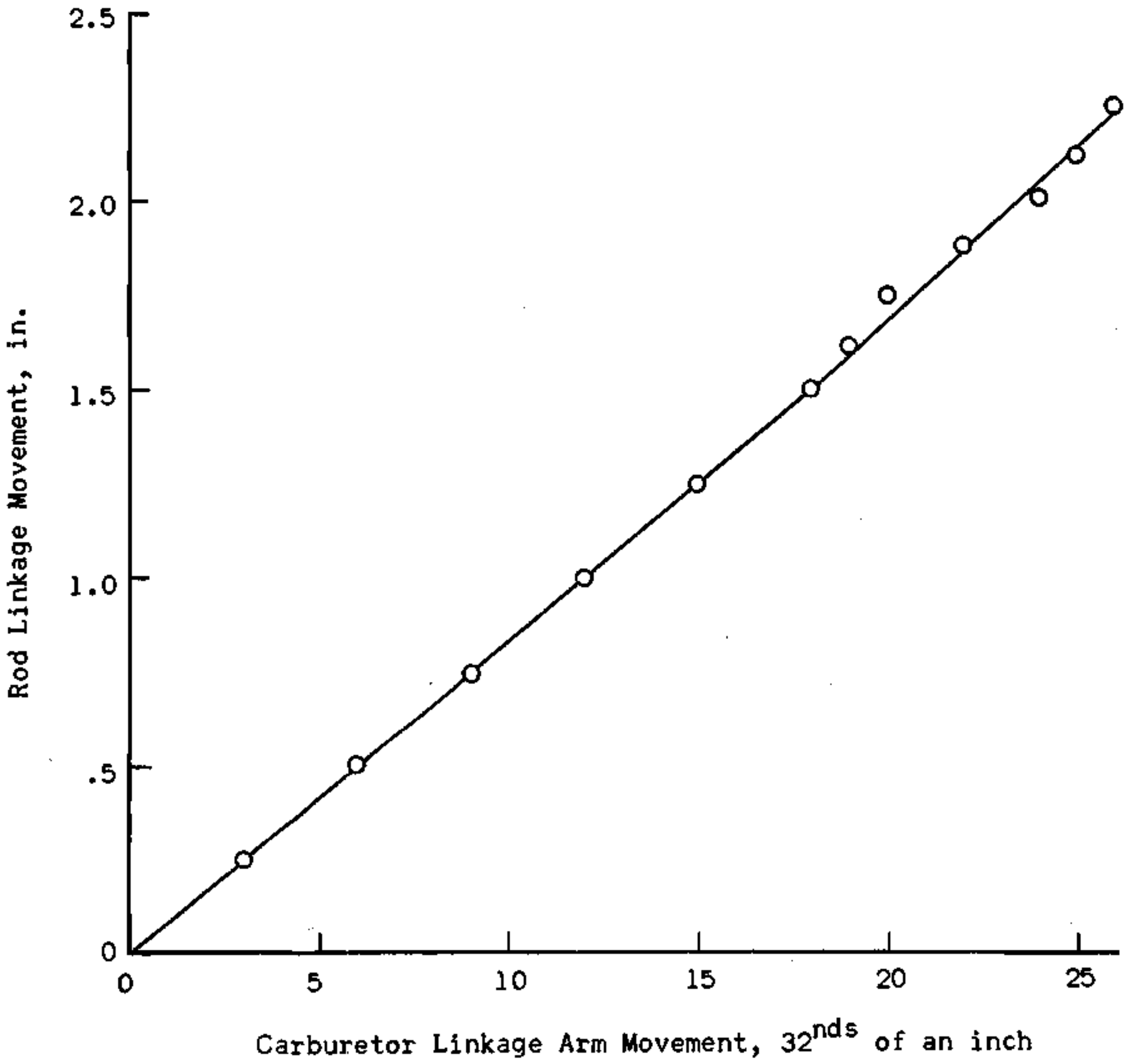

Figure 30. Relationship Between Carburetor Linkage Arm Movement and Extension Rod Movement.

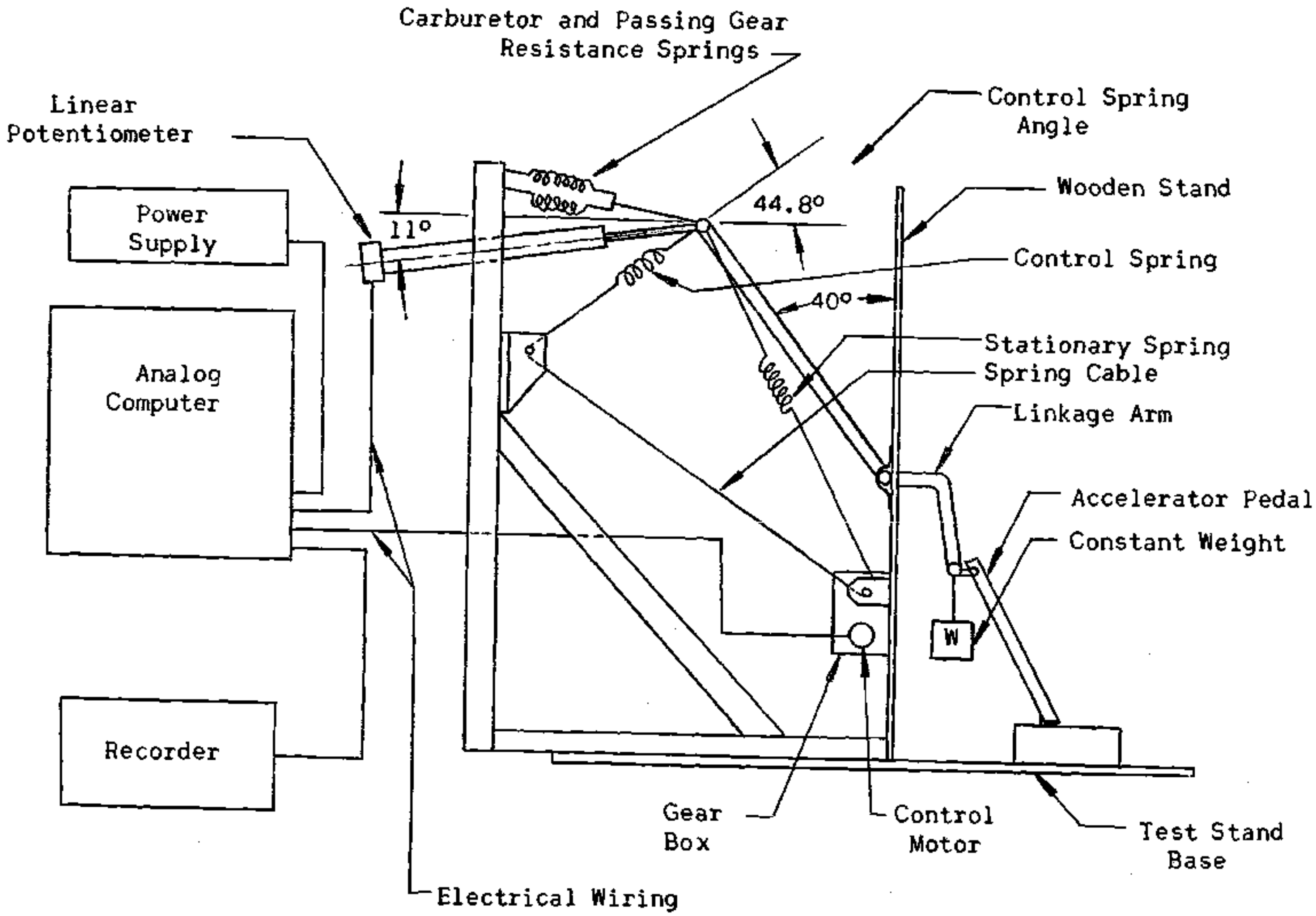

Figure 31. Laboratory Experimental Setup.

linkage arm on the Oldsmobile was disconnected from the main linkage arm (see Figure 29). A small Chatillon spring scale was attached to the carburetor linkage arm. The scale was pulled manually toward the firewall and readings of carburetor drag in lbs. force versus linkage displacement were recorded as shown in Figure 32.

The data of Figure 32 was linearized by drawing straight lines which would best fit the experimental points. Two springs in parallel were used to simulate the carburetor and passing gear resistance on the test stand as shown in Figure 31. One spring had a constant of 3 lbs./in. and was always engaged while the other spring, which had a constant of 10 lbs./in., was only engaged when the linkage had moved 3/4 inch horizontally from its rest position. The reason for the sharp increase in resistance was due to the passing gear engagement which is shown in Figure 32.

A detailed description of the analog computer circuit is given in Appendix III.

A drawing of the control motor, gear box and limit switch arrangement is shown in Figure 33. The motor, gear box, cams, and metal base were secondhand equipment. The 28 volt d.c. motor type 15 MTG-6276-01 was manufactured by the John Oster Co. The drum, drum shaft, shaft supports, and limit switch supports were made in the Mechanical Engineering shop. The limit switches used were Micro type V 3-44.

The drum shaft and the cam shaft turned at a 1 to 1 ratio. This enabled the cams, which were positioned by set screws, to throw the limit switches, which by stopping the motor, limited the rotation of the drum.

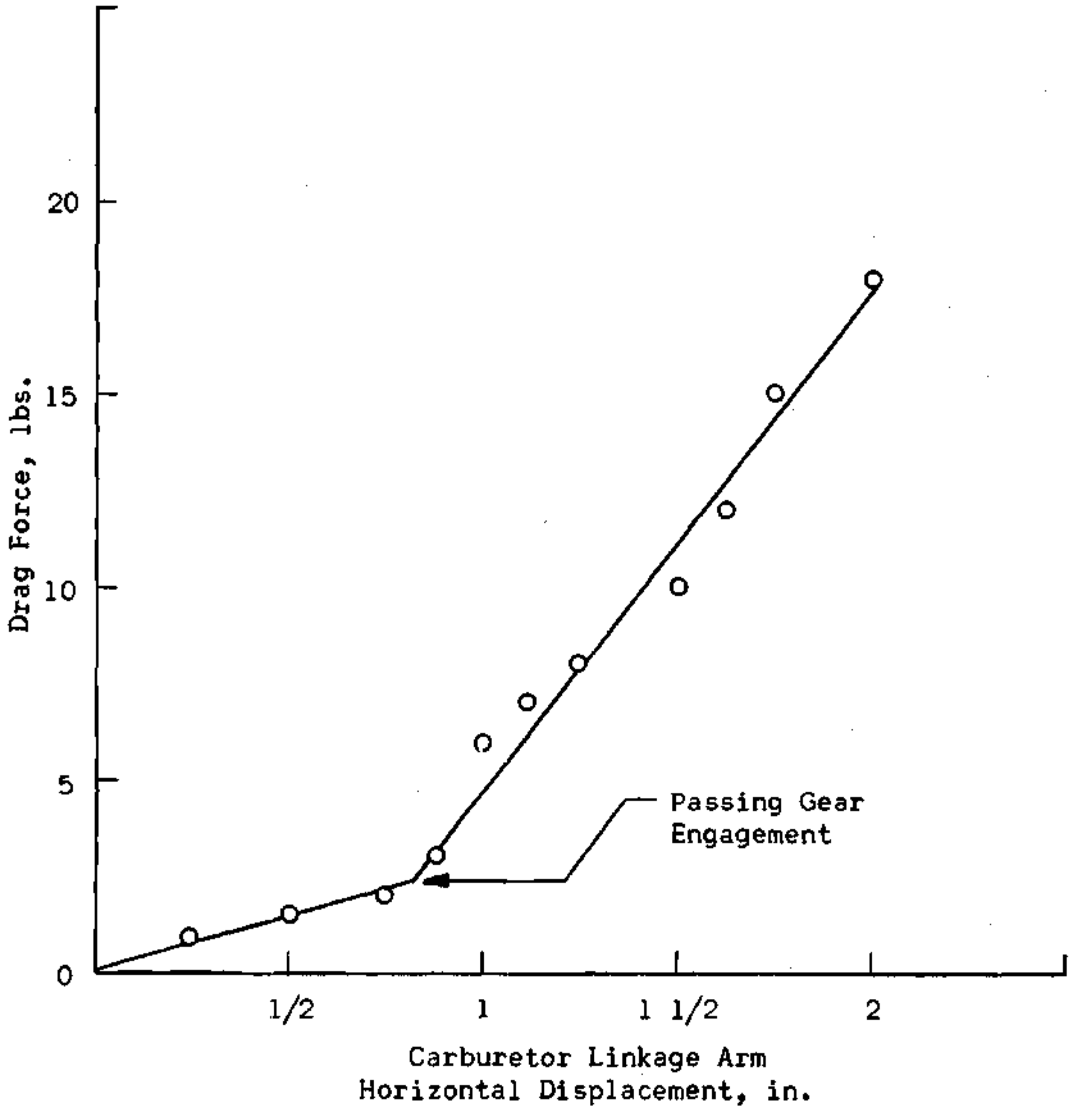

Figure 32. Drag Force of Carburetor and Passing Gear.

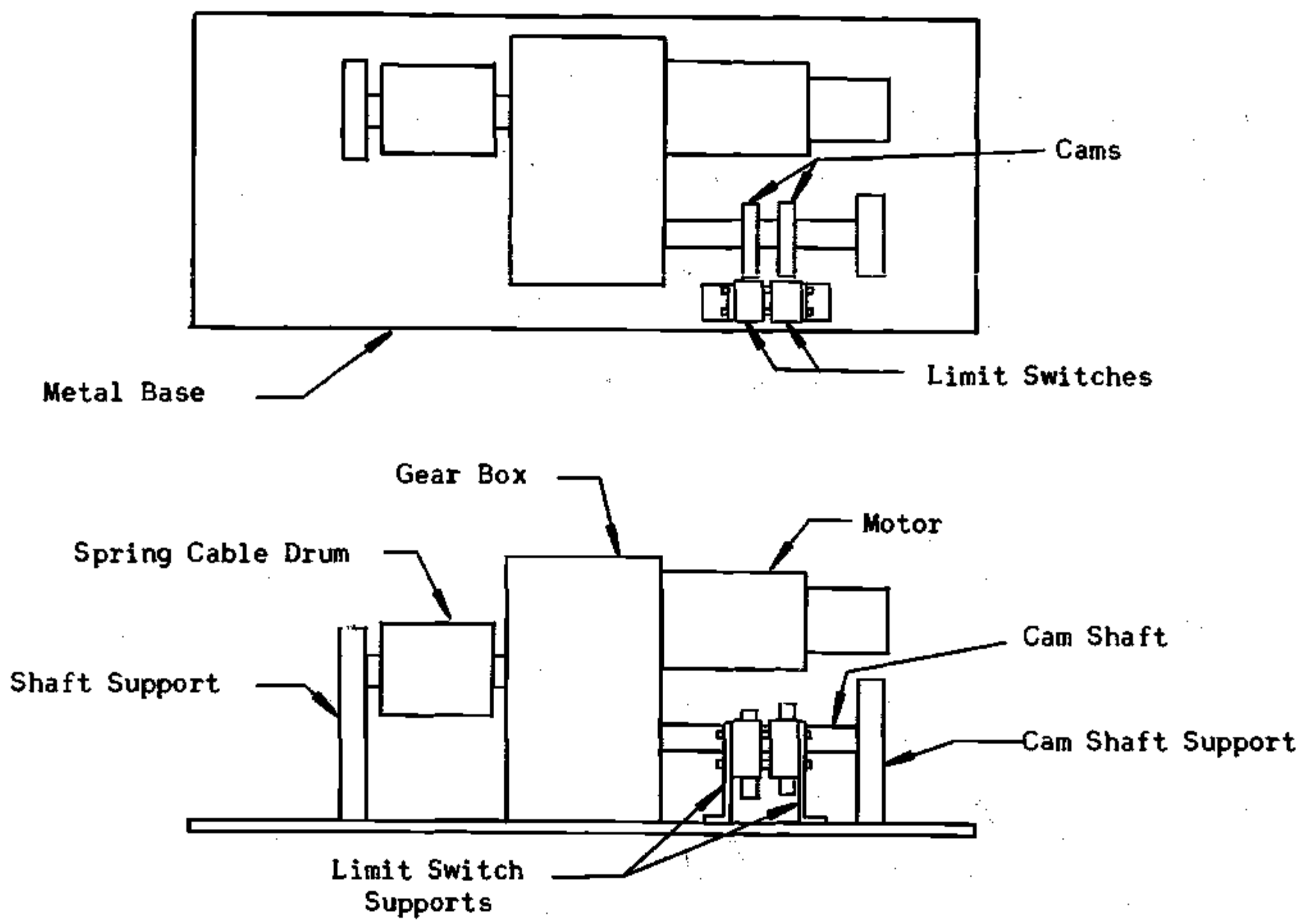

Figure 33. Control Motor, Gear Box, and Limit Switch Arrangement.

The power to run the motor was supplied by a Harrison Laboratories d.c. power supply, model 6204 A. The experimental results were recorded by a Moseley Autograph x-y recorder, model 2D-4.

APPENDIX III

DETERMINATION OF AUTOMOBILE AND CONTROL MOTOR TRANSFER FUNCTIONS

Determination of Mathematical Model for the Automobile and the Related Analog Computer Circuits

The results of the automobile acceleration and deceleration tests are shown in Figures 34 and 35 respectively. By inspection of Figures 34 and 35 it can be seen that all the curves closely resemble the step response of an overdamped second-order system. The transfer function for the classical second-order system is

$$\frac{C(S)}{R(S)} = \frac{\omega_n^2}{S^2 + 2\xi\,\omega_n S + \omega_n^2}$$

where

S = Laplace Operator

$C(S)$ = Output of System

$R(S)$ = Input of System

ω_n = Natural Frequency of System

ξ = Damping Ratio of System

Since speed control systems are mainly operated at high speeds for highway operation, it was decided to simulate the curves which had a starting velocity of 55 mph. The second-order system was simulated on the Systron Donner (Model SD 10/20) analog computer as shown in Figure 36. By trial and error the gains for integrator 1 which gave the best

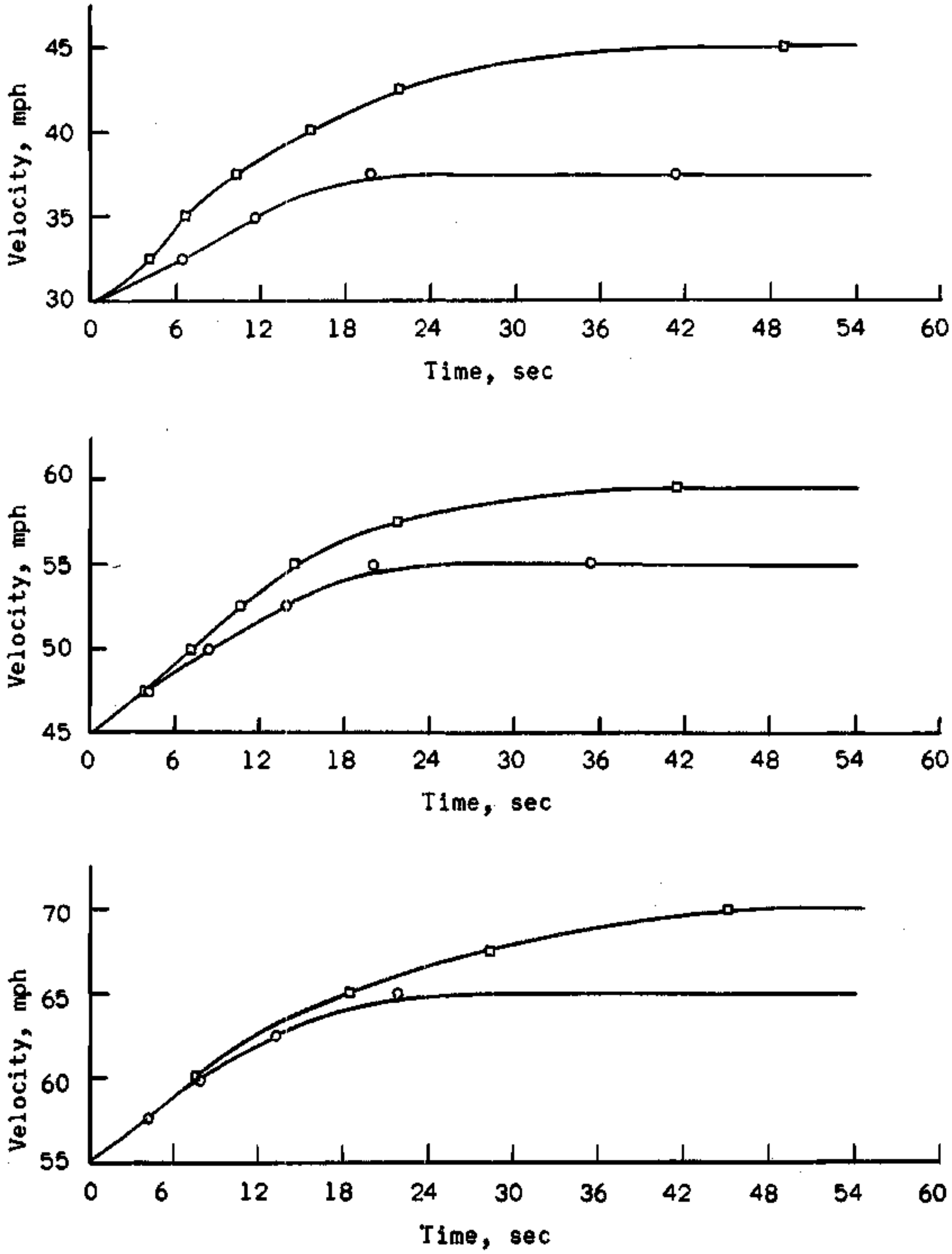

Figure 34. Velocity Responses of the Oldsmobile to Positive Steps of Input to the Accelerator Pedal.

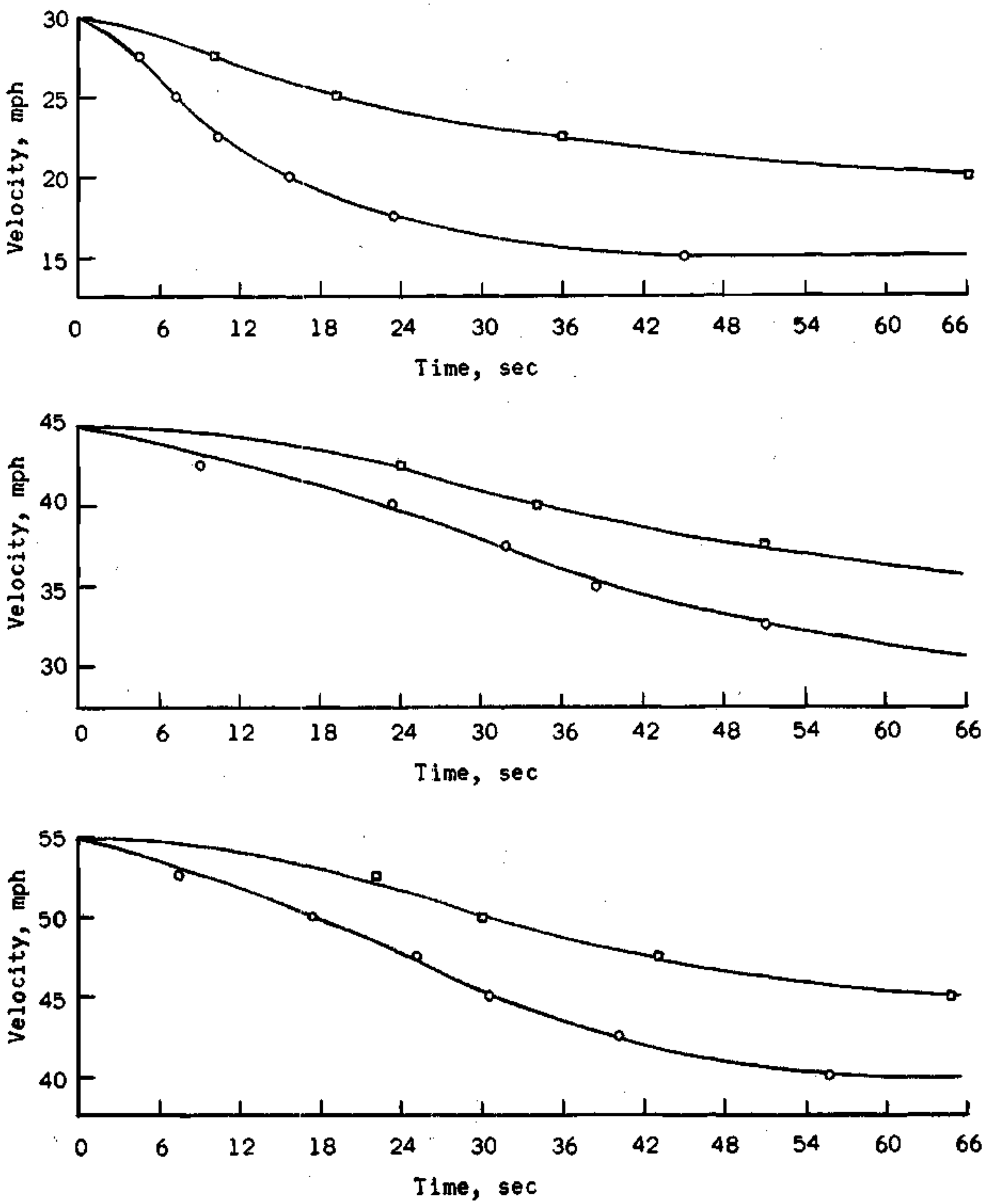

Figure 35. Velocity Responses of the Oldsmobile to Negative Steps of Input to the Accelerator Pedal.

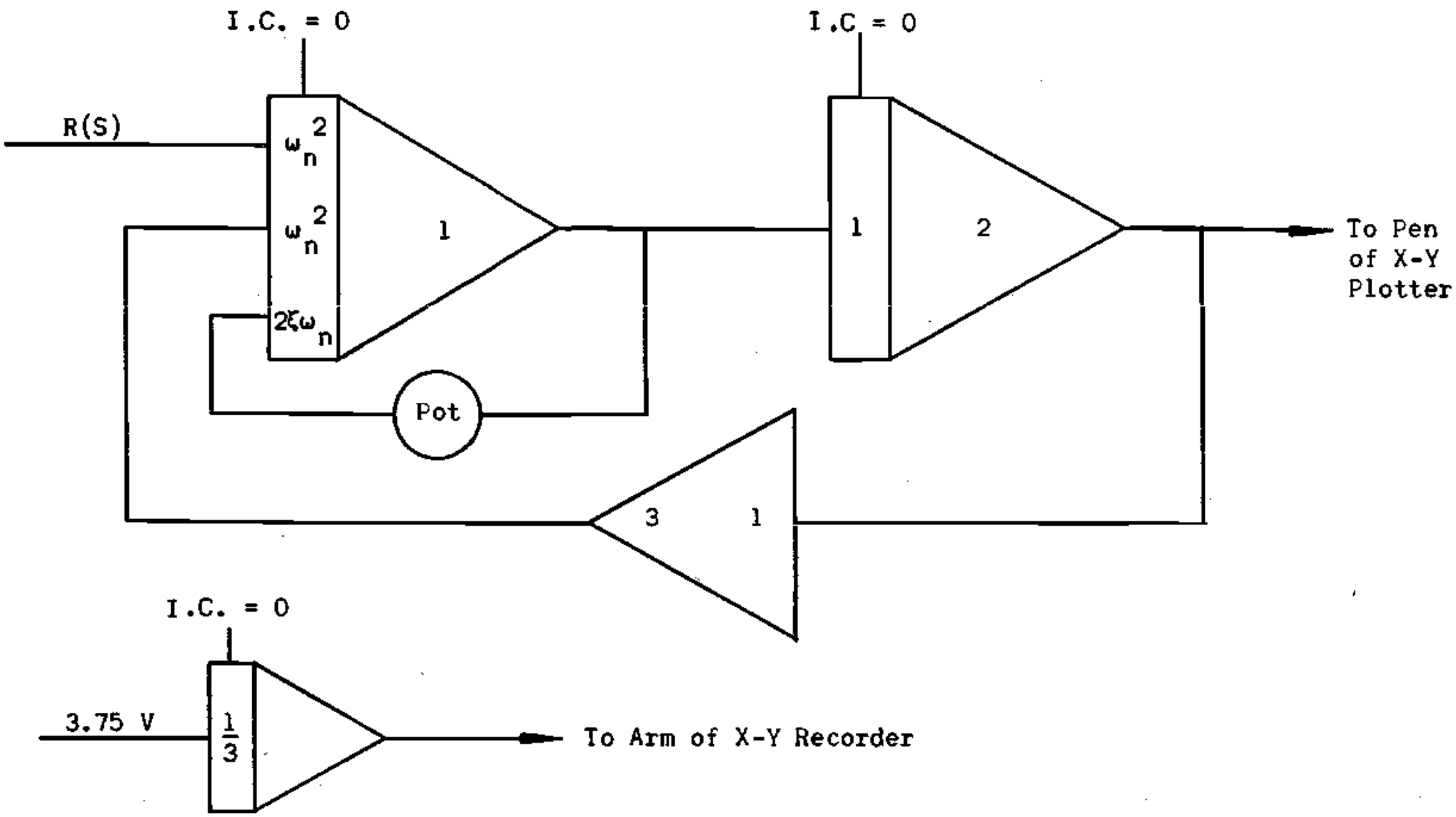

Figure 36. Analog Computer Simulation of Second-Order System and Time Base.

simulation of the experimental data were found to be

$$\omega_n^2 = 5$$

$$2\zeta\omega_n = 100$$

The results of the simulation were plotted by the X-Y recorder and compared with the experimental results in Figure 37.

The complete system electrical circuit diagram used for the experimental tests is shown in Figure 38. Because the Systron Donner analog computer only had 6 amplifiers, a Dymec amplifier (model 24-60A) was used to complete the system requirement of 7 amplifiers.

Determination of Control Motor Transfer Function

In order to make the analytical analysis a transfer function had to be determined relating the output voltage of the linear potentiometer, which indicated the position of the accelerator pedal, to the input voltage to the control motor. During the motor engagement time the potentiometer voltage changed 7.5 volts in 2.4 seconds. This corresponded to a motion of .67 in./sec of the control spring cable and a motion of .26 in./sec of the linear potentiometer. Assuming the change to be linear this amounted to a voltage change of 3.1 volts/sec. When this was fed to the computer, it was multiplied by a gain of 10. The resulting motor transfer function was 31/S.

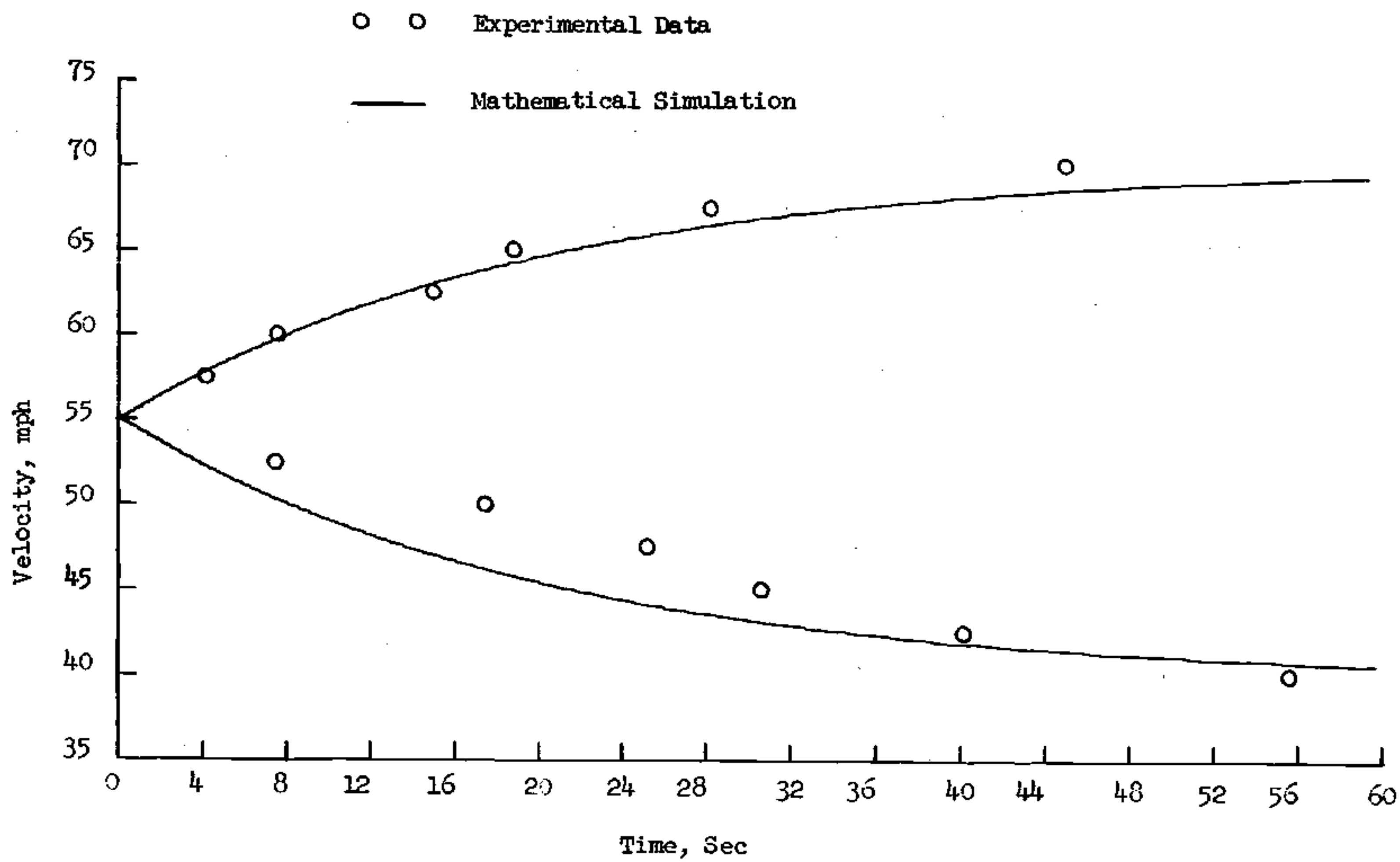

Figure 37. Comparison of Mathematical Simulation to Experimental Data.

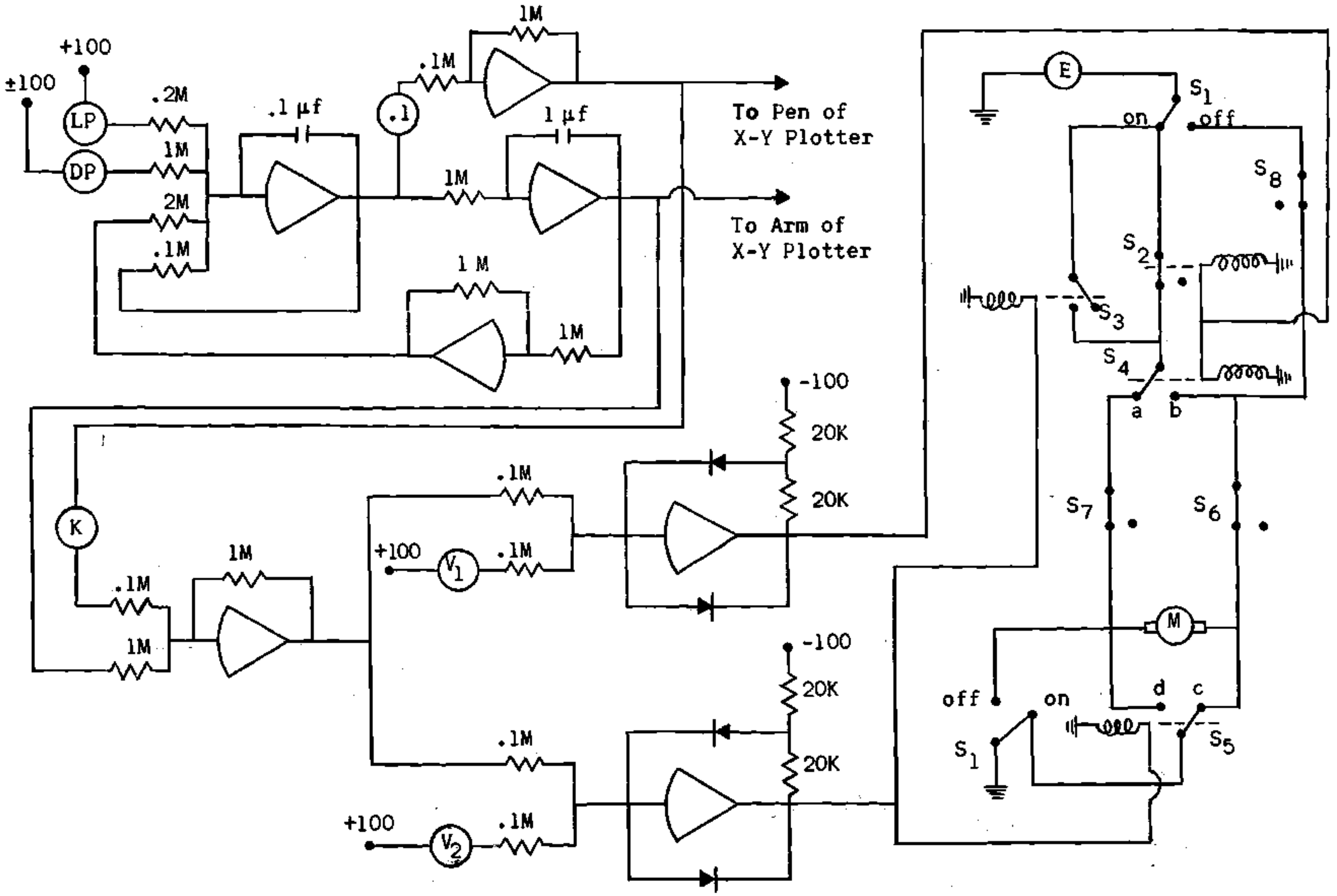

Figure 38. Electrical Circuit Diagram of Experimental Setup.

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3,099,330

3,114,427

3,268,026

3,268,027